VENTILATION CONTROL FOR ENERGY CONSERVATION: DIGITALLY CONTROLLED TERMINAL BOXES AND VARIABLE SPEED DRIVES

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To Rob Knapp,
who got me into this mess in the first place

ABSTRACT

For several years, electronic variable speed drives (VSDs) have been used on fan motors in large building variable air volume (VAV) ventilation systems, often replacing inefficient variable inlet vanes (VIVs) as a means of regulating supply duct pressure and return airflow. Few, if any, published studies of measured energy savings and fan performance have been done. The first major portion of this thesis is a case study of a VSD retrofit, in which the drives were installed on two 50 hp and two 20 hp fan motors in a commercial office building with a variable air volume distribution system. Pre- and post-retrofit part-load fan motor performance is examined using analysis of motor input power as a function of air flow. This analysis shows that despite considerable savings over inlet vanes (35%), their high cost results in a relatively long payback period, favoring incorporation of VSDs in new construction rather than as a retrofit item, except in buildings operating continuously, for which the payback is 2.7 years. Unusually high pre- and post-retrofit flow rates and energy consumption for one of the fans were found to be the result of a distribution system malfunction. The significant reduction in savings underscores the necessity of correcting distribution system problems as part of any VSD retrofit.

Measurements revealed that reducing supply duct static pressure can significantly boost the savings achievable with VSDs. This is not ordinarily possible, however, in conventional VAV systems with pneumatic zone control, where duct static pressure must be held constant at a high level so that terminal boxes are not "starved" for air during high cooling load conditions. Decreasing static pressure in a nearly identical system employing VIV fan control had no significant effect on fan power, in contrast to the results for VSDs, where the magnitude of savings due to decreasing static pressure set point is half of the savings due to the installation of VSDs alone.

The second major focus of this thesis is on terminal boxes using direct digital control (DDC). DDC has only recently been applied to terminal boxes, and has opened a new realm of possibilities for improving supply fan and zone temperature control, both in energy-efficiency and comfort; nevertheless, little field experimentation with new control strategies has been done. DDC terminal boxes have two advantages over their pneumatically controlled predecessors that make them particularly well-suited for improving fan control in VAV systems: inherent communications capability and improved flow characteristics—supply air flow rate is controlled to set point, independent of pressure. In a system using VSDs to regulate duct static pressure, these two features would permit a supply fan control strategy in which duct static pressure can be minimized, without sacrificing occupant comfort or adequate ventilation.

In the course of developing a static pressure minimization control strategy, it was necessary to explore the response of terminal boxes to changes in inlet pressure. An instrumented DDC terminal box was installed in an unoccupied office space; test results were used to create and calibrate computer models of a DDC fan-powered terminal box, ducts and zone, for use with the public domain dynamic simulation program *HVACSIM+*. Zone temperature and flow rate control loops are represented using a discrete-time proportional plus integral controller model. Validation of the system model was performed by subjecting the model to time-varying boundary conditions derived from measured data, with satisfactory results.

Preliminary evaluation of the proposed static pressure minimization control algorithms using simulation yielded encouraging results, showing that such a strategy is feasible in systems combining VSDs and DDC terminal boxes. Other control improvements possible with DDC zone control technology are discussed. Finally, an examination of the current status and future directions of dynamic HVAC simulation tools is presented.

Acknowledgements

Writing a thesis is, by necessity, a self-centered activity. At the same time, the researcher depends on others for their advice, knowledge, critique, and support. Combined, these two factors make for a one-way flow of energy, so to speak. It is not until the final document is in hand, that the author has the feeling of "giving something back."

Indeed, I hope this work contributes something to a field that cannot, these days, have too much help. The earth is telling us in a loud voice that we must change the way we live and use resources. It is this voice that has inspired my work, my reply to its call. As we formulate our reply as a society, as a species, energy efficiency will play an essential role. We must renew support for experimentation in this area. We must try, try, try, as many solutions as possible, with careful study of what works and why.

I thank those without whom this project would have been impossible, who also hear the voice, and answer it as well:

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This thesis carries '1885-T' in the records of the Department of Mechanical and Aerospace Engineering.

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Introduction

1.1 AIR HANDLING ENERGY USE IN COMMERCIAL BUILDINGS

Commercial buildings[1] in the U.S. are estimated to use 1,480 TWh of energy per year (DOE/EIA, 1986b), of which 48%, or 710 TWh (2,340 TWh_{th})[2] is electricity. About 72% of this electricity is produced using fossil fuels, 20% is nuclear, and most of the remainder is hydroelectric. (DOE/EIA, 1989). While the need to reduce energy expenditures, and the high cost (both monetary and environmental) of constructing new generating capacity have been forces motivating conservation in the last two decades, an increasing awareness of the contribution of fossil fuel combustion to global warming has renewed interest in making buildings more energy efficient.

In commercial buildings, ventilation accounts for about 10% of the electrical energy used (**Figure 1.1**), or about 64,000 GWh/yr (EPRI, 1986). This costs consumers $4.3 billion per year,[3] and amounts to 2.5% of the total annual electricity generation in the U.S.

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[1]Buildings are usually classified, on the basis of primary use, into three groups: residential, commercial, and industrial. The commercial category (as defined by DOE/EIA, 1986a), in general, includes all buildings that are less than 50% residential, and less than 50% manufacturing (by floor area).
[2]Thermal energy, assuming transmission loss of 10%, and an average generation efficiency of 33%.
[3]Using the average cost of electricity used in commercial buildings in 1986, $0.0673/kWh (DOE/EIA, 1986b).

for 1988 (DOE/EIA, 1989). The generating capacity required to provide this electricity is about 19 GW,[4] or 57 GW_{th}.[5] Thus fans are a prime target for conservation.

ENERGY USE IN VAV SYSTEMS

Of the several types of building heating and cooling systems using ducted forced air, variable air volume (VAV) systems are relatively common, used in about 22% (by floor area) of all commercial buildings, and 41% of commercial buildings that are both heated and cooled (DOE/EIA, 1986a).[6] Thus, the energy used by fans in VAV systems alone is 14,000 GWh/yr. VAV systems are so-called because the supply air volume, rather than

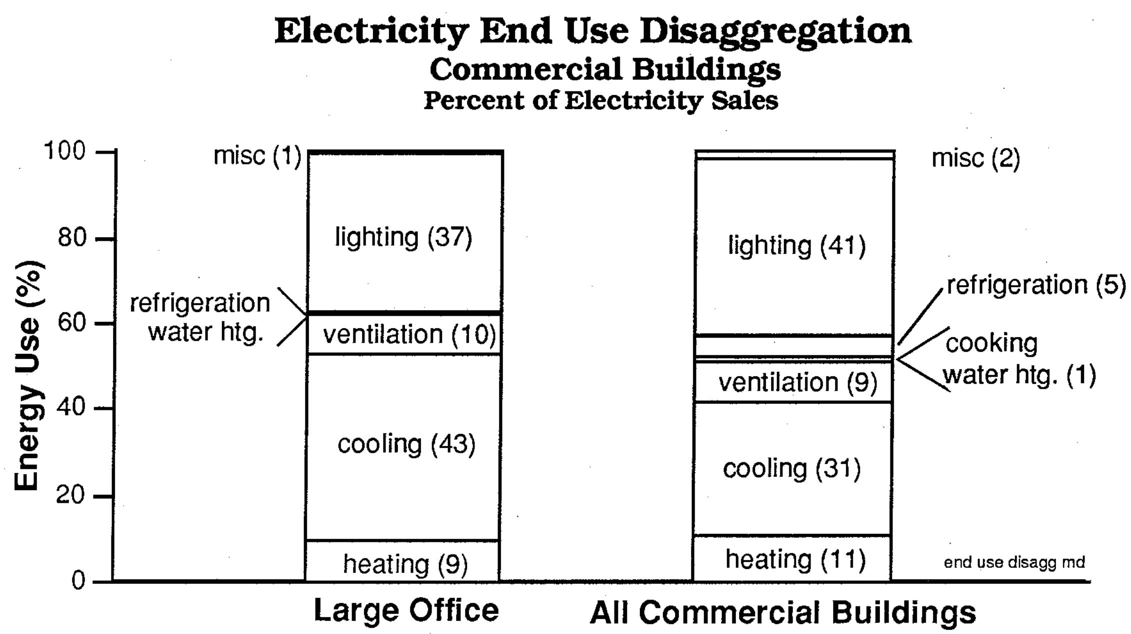

Figure 1.1. Disaggregation of commercial building end use electricity, by share of electricity sales. Source of data: Electric Power Research Institute (1986). "The COMMEND planning system: National and regional data and analysis." EPRI EM-4486. Palo Alto, California.

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[4]Using an area-weighted average of 71 hours per week of operation (DOE/EIA, 1986a), and assuming a transmission loss of 10%. A typical nuclear plant produces 1 GW.

[5]Assuming an average generation efficiency of 33%.

[6]It is often suggested that VAV systems are more common in newer buildings. The NBECS data compiled through 1986 by DOE/EIA do not show this to be true for buildings constructed prior to that year.

temperature, is varied as the space heating or cooling load changes. They are known for providing relatively good control of space temperatures, and using comparatively less fan energy than constant volume types (Rickelton and Becker, 1972).

In general, a VAV system consists of a supply fan, a return fan, and terminal boxes in each "zone" of the conditioned space (**Figure 1.2**). A zone, served by a single terminal box and thermostat, may consist of several rooms. If the air is supplied at a constant temperature (usually around 13 °C or 55 °F), a terminal box controls the volume of air going into a space so as to regulate the space temperature at a value set at a local thermostat. As the flow through the boxes changes, the pressure developed by the supply fan is controlled so that static pressure in the supply duct is held constant. Variable inlet vanes (VIV)—dampers at the fan inlet—have traditionally been used for this purpose and to control the

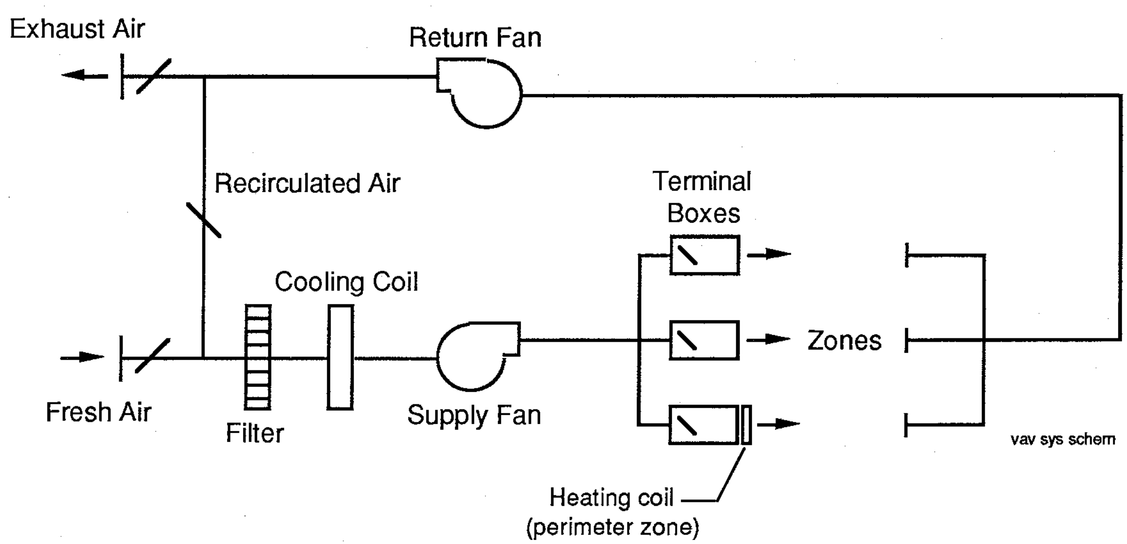

Figure 1.2. *Schematic diagram of basic VAV system. Cool air is supplied to the terminal boxes at a constant temperature; terminal boxes vary the flow as necessary to cool individual spaces, or zones. During the heating season, the cooling coil is off, and terminal boxes add heat locally as required. Terminal boxes may use built-in fans to recirculate room air and provide a constant outlet volume, or the outlet volume may vary, with no recirculation. The return fan tracks the flow of the supply fan to maintain a fixed fresh air flow rate. A large building may have one or more VAV systems serving many zones.*

return air flow rate; recently variable speed drives (VSDs)[7]—electronic devices which control the speed of the fan motor—have gained in popularity as a more efficient means of controlling pressure and flow.

DISAGGREGATING AIR HANDLING ENERGY USE

To understand why air handling comprises such a substantial portion of a commercial building's electricity use, it is useful to disaggregate energy entering a fan motor or drive into component losses. The building used experimentally in this study serves as a good example. For now, we'll consider only a single supply air handling system serving one half of the building; detailed descriptions of the building itself will follow in later sections.

In order to provide fresh air to the conditioned spaces in our building (many of which are in interior areas), we must somehow move it from the outside to these spaces. This is done with a single distribution system, as shown in **Figure 1.2**. This air also serves as a vehicle to carry heat to or away from the spaces. The amount of air required for ventilation is sufficient for heating, but is not enough for cooling.[8] On a summer day (ambient temperature 34 °C, 93 °F), the peak sensible cooling load[9] for half of the building can be estimated[10] as 245 kW, of which 120 kW is for lights and occupant equipment,[11] 60 kW is for (sensible) cooling of fresh air,[12] 55 kW is for conduction and infiltration gains,[13] and 10 kW is for solar gain through glazing.[14] The volumetric rate of air flow required to remove 245 kW if the supply air is 13 °C (55 °F) and interior temperature is 23 °C (73 °F), is:

[7]Also called adjustable speed drives (ASDs).
[8]Even during the heating season, however, flow rates in this building are higher than the ventilation minimum, as interior zones require cooling.
[9]Latent cooling loads, which affect chiller energy use, do not influence the amount of air required for cooling.
[10]Unless stated, basis of cooling load estimates is Hsieh (1988).
[11]120 kW = (20 W/m^2)(6,000 m^2).
[12]60 kW = (4.7 m^3/s)(11 °C)(1.1 kg/m^3)(1.0 kJ/kg-°C)
[13]55 kW = (10 kW/°C)(34 °C – 23 °C)(0.5 building);
[14]10 kW = (220 W/m^2)(90 m^2)(0.5); South glass, 2 pm.

$$\frac{245\ \text{kW}}{(23\ °\text{C} - 13\ °\text{C})(1.1\ \text{kg}/\text{m}^3)(1.0\ \text{kJ}/\text{kg-}°\text{C})} \approx 22\text{m}^3/\text{s}\ (47{,}000\ \text{cfm})$$

This flow rate is, in fact, in the neighborhood of flows observed in this building during periods of high cooling load.

The equipment used to move this air converts electrical energy to kinetic energy—by raising the velocity of the air—and potential energy, in the form of static pressure. This conversion of energy is far from 100% efficient, due to losses at each step of the process: energy is lost due to the efficiency of the motor, friction in the transmission and fan shaft, and turbulence in the outgoing air stream. For now, however, let us consider an ideal fan, whose input power P (in kW) is equal to the product of the volumetric flow rate $\dot{V}$ (in m^3/s) and the total pressure rise in kPa across the fan Δp_t:

$$P = \dot{V}\Delta p_t \tag{1.1.1}$$

The total pressure term can be broken into two components, static and dynamic:

$$\Delta p_t = \Delta p_s + \Delta p_d \tag{1.1.2}$$

The dynamic pressure difference Δp_d is proportional to the square of the exit velocity v, since at the location just upstream of the fan where pressure is measured, the velocity is small and can be ignored:

$$\Delta p_d = \frac{1}{2}\rho v^2 \tag{1.1.3}$$

The density of air ρ will be assumed to be constant, at 1.1 kg/m^3. The velocity is equal to the flow rate divided by the duct area at the location in the duct downstream of the fan where pressure and velocity are measured, about 2.2 m^2.

FULL LOAD CONDITION

Motor and Fan Losses

Duct velocity is typically below 13 m/s (2500 fpm) to keep duct noise at acceptable levels. In the study building, at full flow (22.2 m^3/s), this velocity is 10 m/s (1960 FPM). The dynamic pressure difference at this flow rate is then about 55 Pa (0.22 in. H_2O). The static pressure across the fan is equal to the total pressure minus 55 Pa, and can be measured directly, or determined (given fan speed and flow rate) from the manufacturer's graph for this fan, as 1250 Pa (5.0 in. H_2O). Thus, dynamic pressure is less than 5% of the total pressure rise, and most of the fan power goes into increasing the static pressure.

The full load power required by an ideal fan in this building would be (22.2 m3/s)(1305 Pa) = 29 kW. The electrical power actually used by the fan under these conditions has been measured as 42 kW, giving a total efficiency of 29/42 = 0.69. Fan/motor losses[15] are thus considerable, and it is therefore useful to estimate how they are allocated among the loss components mentioned above.

Motor. The motor (50 hp) has a nameplate efficiency of about 0.90; although this varies as a function of load, the variation is relatively flat above 45% of full load, so we will use this value. The motor thus accounts for 0.1 / (1 – 0.69) = 32% of the total fan/motor loss.

Fan. From the manufacturer's graph, we can read, at this operating point, a value for shaft input power of about 36 kW. The 7 kW loss in the fan is due to aerodynamic resistance in the fan itself, bearing friction in the fan shaft, and turbulence in the exit air stream. The ratio of fan output power to shaft input power is referred to as the fan efficiency, which here is 29 kW / 36 kW, or about 0.81. The fan thus accounts for 7 / (42 – 29) = 54% of the total.

[15]The term "losses" is used here because the "missing" energy is no longer available for useful work in getting the air to where it's going.

Transmission. The only source of loss between the motor and the shaft is the transmission, consisting of pulleys and belts; the transmission efficiency can be calculated as the ratio of fan input power to motor output power, or 36 kW / (42 kW x 0.90) = 0.95, and accounts for (1 – 0.95) / (1 – 0.69) = 16% of the total loss.[16]

Since all of the losses occur in the air stream, 13 kW of heat is added to the flow, increasing its temperature by 0.53 °C.

The major portion of the full load fan/motor loss is in the fan. Can this be reduced? This fan is a backward-inclined bladed type, which along with backward-curved bladed and airfoil fans, is the most efficient of the several classes of fans used in HVAC applications. The next largest loss, the motor, is a potential area for improvement (i.e., by replacement with a more efficient type); the efficiency, however, is already 90% and increasing it by 5% would have relatively little benefit. Since motor technology is beyond the scope of this study, improving motor efficiency will not be explored further here.

Distribution and Other Losses

What about reducing fan power, i.e., by reducing flow or static pressure? We can decrease flow and maintain cooling capacity only by reducing the temperature of the supply air. Doing so will increase chiller energy use, however, perhaps by more than we save at the fan. The energy tradeoff between supply air temperature and flow is a complex issue, and while beyond the scope of this study, deserves further research. The effect of supply air temperature on energy use in this building has been given a preliminary exploration in work by Norford et al. (1986).

Before considering the reduction of fan outlet pressure, we must look at the fan in the context of the whole air distribution system. So far, we have accounted for 31% of the useful work done by the electricity input—as electrical, mechanical, and aerodynamic

[16]The sum of the three percentage loss figures is slightly more than 100% because of round-off.

losses in the fan and motor. The rest is lost primarily in overcoming resistance to flow, mostly between the fan and the zones, some upstream of the fan. Graham (1989) has disaggregated the input energy into estimated loss components, based on measurements in a typical multi-level commercial office building (**Figure 1.3**). Here, the fan and motor account for only 25% of the input electricity—somewhat less than our estimate, perhaps because ours is calculated at full load, when throttling losses are lower.[17] The figure also shows distribution ducting accounting for about 60% of the total input, and 20% for losses upstream of the fan.

Let's take a closer look at the component losses for the study building. **Figure 1.2** shows fresh air (at atmospheric pressure) entering from outside the building and being mixed with return air. In the study building, this intake is at basement level, where the fans are located. The air then passes through a filter for dust removal, in order to improve air quality and reduce fouling of cooling and heating coils. The air then passes through a cooling coil[18] into a small "room" where the fan is located. The pressure in the fan room is below atmospheric, perhaps –150 Pa (–0.60 in. H_2O). After leaving the fan (at 1.1 kPa, or 4.4 in. H_2O under full load), the air is pushed through a main trunk which soon becomes vertical and rises to the third floor, with take-offs at the lower two floors. The structure of the distribution duct work is tree-like, with ducts getting smaller as they branch off. At the end of the supply, or primary ducts, are terminal boxes which throttle the flow according to temperatures measured by thermostats located in the spaces served by each box. The air is then ducted from each terminal box to one or several rooms, which it enters through diffusers in the ceilings.

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[17]Throttling losses are discussed below. It is unclear whether return fans are included in Graham's estimate—not all large commercial buildings have them.

[18]A bypass damper allows much of the air to go around the cooling coil when it is not in use.

In VAV systems, ducts are usually designed so that the static pressure at the inlet of the farthest terminal box on a floor is above some minimum required by the box to produce the necessary flow. This minimum depends on many factors particular to the terminal box and installation, but let us assume a typical value here of 0.125 kPa (0.5 in. H_2O) (relative

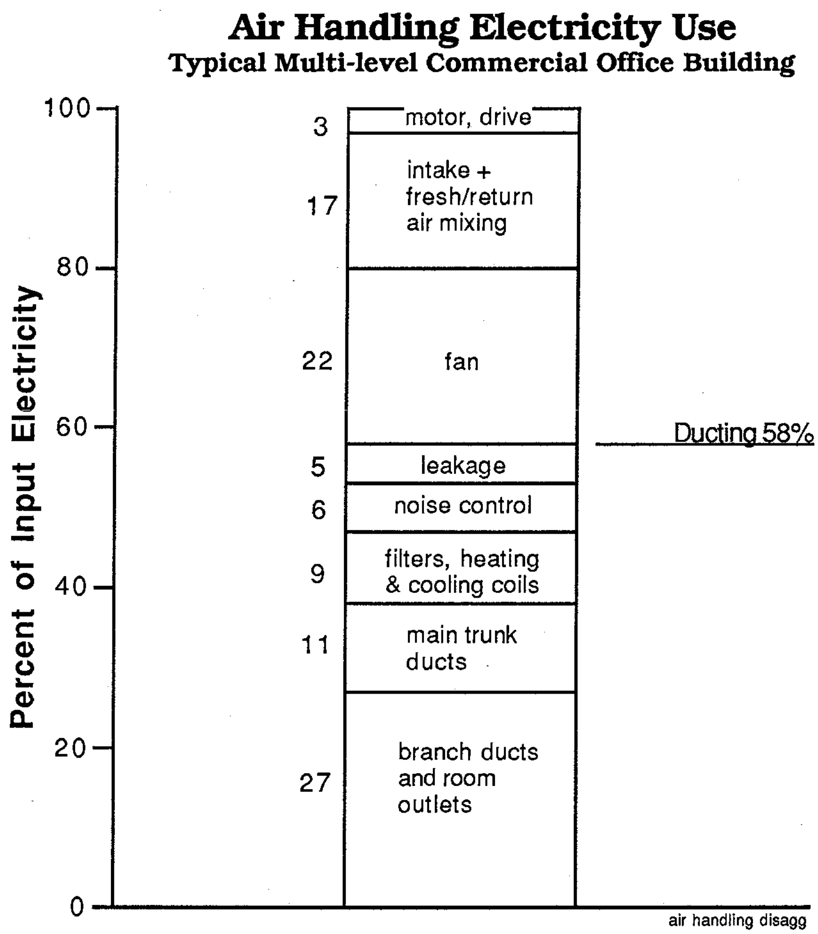

Figure 1.3. Disaggregation of energy used to move air into component losses, based on measurements made in a multi-level commercial office building in the U.S. All of the useful work, input as electricity, goes to overcome mechanical and aerodynamic losses in the motor, fan, and ducts. Source: Graham, J. B., "Air Handling," Technology Menu for Efficient End-Use of Energy. *Vol. 1. Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.*

to the local room pressure)[19] for the farthest box on the third floor. Also required, in order to lessen noise, is that the pressure at the inlet of the closest box does not go above some maximum, typically 0.37 kPa (1.5 in. H_2O). The static pressure at the fan outlet is kept as high as it is in order to satisfy terminal box minimum inlet pressure conditions at all times.[20]

Let us assume that the air velocity in the duct upstream of the terminal box is the same as that leaving the fan. This is a reasonable assumption, and it enables us to ignore dynamic pressure, which is small anyway. To recap, a static pressure of 0.125 kPa is required at the supply duct terminus, and 1.25 kPa (5.0 in. H_2O) is produced by the fan. This amounts to a loss of 1.13 kPa (4.5 in. H_2O).[21] According to ASHRAE (1982), the static pressure losses in a system can be represented as the sum of losses in the ducts, coils, and filters:

$$\Delta p_{ls} = k_d \dot{V}^2 + k_c \dot{V}^c + k_f \dot{V}^f \tag{1.1.4}$$

where

Δp_{ls}	=	total static pressure loss (Pa)
k_d	=	duct flow resistance coefficient (m^{-6})
k_c	=	coil flow resistance coefficient (m^{-6})
c	=	coil flow exponent, between 1.46 and 1.81
k_f	=	filter flow resistance coefficient (m^{-6})
f	=	filter flow exponent, between 1.01 and 1.79, for filter efficiencies of 35% to 95%.

[19]This building is slightly pressurized (via return fan flow modulation) with respect to the outside, so that exhaust systems work properly, and also to reduce infiltration. The pressure difference is small (<25 Pa) as is typical for buildings of this type (The Trane Company, 1982b) and is therefore ignored in this simple analysis.

[20]It is actually the pressure at a point in the duct between the fan and terminal boxes that is controlled; more detail on this will follow.

[21]In order to overcome gravity in pumping the air 15.2 m (50 ft) from the basement to the third floor, (1.1 kg/m^3)(9.8 m/s^2)(15.2 m) = 0.164 kPa (0.66 in. H_2O) is required. We implicitly account for this component by using a relative, rather than absolute pressure at the terminal box inlet.

The duct loss term is a form of the familiar equation for turbulent flow in pipes, with the assumption of constant density, and combines frictional and fitting losses. The terms for coils and filters each contain a laminar and a turbulent component, hence the exponents between 1 and 2. For the full load condition, these two terms are easily estimated using measured data and design specifications.

Coil. The cooling coil is specified to have a pressure drop of 0.125 kPa (0.5 in. H_2O) at 28.3 m^3/s (60,000 CFM). Using Equation 1.1.4, we can estimate k_c as 0.59, taking an intermediate value of 1.6 for the flow exponent. For a flow of 22.2 m^3/s, the pressure loss due to the cooling coil is thus 85 Pa (0.34 in. H_2O).

Filters. The filters are specified to have a pressure drop of from 75 to 250 Pa (0.3 to 1.0 in. H_2O) for a flow of 28.3 m^3/s and a filter efficiency of 0.80. Using a procedure similar to that for the coil, and assuming an intermediate filter loading, the full load filter pressure drop can be estimated as 100 Pa (0.4 in. H_2O).

Intake. The loss due to the outside air intake is part of the duct component in Equation 1.1.4. Here, however, we wish to group the losses upstream of the fan, so this will be considered separately. Summing the coil and filter losses, we get 185 Pa, in the same neighborhood as the typical value used above for the pressure difference between the fan inlet and the atmosphere, 0.15 kPa. We have no simple way to estimate directly the loss due to the intake; rounding 0.185 up to 0.2 kPa (0.80 in. H_2O) should account for this, approximately.

Duct. Subtracting 0.2 kPa from the static pressure difference between the fan and the terminal box, 1.13 kPa, yields the portion attributable to frictional losses and dynamic losses (resulting from disturbances in the flow at fittings) in the distribution ducts: 0.93 kPa (3.7 in. H_2O).

In summary, we have disaggregated the full load fan static pressure into three components:

0.93 kPa	74%	distribution duct losses
0.20 kPa	16%	intake, filters, coil
0.13 kPa	10%	terminal box and zone ducting

We can also further break down motor input energy:

51%	distribution duct losses
31%	fan, motor, transmission
11%	intake, filters, coil
7%	terminal box and zone ducting

Distribution duct losses are easily the largest component. Both Graham (1989) and Tsal (1986) recognize this as a problem for large buildings in general, and attribute it largely to the use of duct design techniques which are not able to optimize for energy or life-cycle cost. Designers generally weigh the factors of noise (which increases with velocity), capital cost (which increases with duct size), and energy costs (increasing with higher static pressure requirements). According to Tsal, three requirements must be met simultaneously in an optimal duct design: 1) the fan must operate at optimum system pressure, 2) the optimal ratio of velocities for all duct sections must be selected, and 3) pressures must be balanced through changes in duct sizes rather than with dampers or other obstructions. While the last requirement is attainable with currently used design tools, the most common methods—equal friction, static regain, velocity reduction, and constant velocity—cannot (inherently) satisfy the first two conditions. A relatively new (10 years old) technique—the T-Method—optimizes the duct design by minimizing an objective function, using tee-staging in a manner similar to dynamic programming, but with local optimization at each stage (Tsal et al., 1988). Applying this method to typical systems has been shown by Tsal and Behls (1989) to minimize life-cycle system cost and result in electricity savings of 20 to 40% over conventionally designed systems.[22] The new method, however, requires a microcomputer, and available software is limited in the complexity of systems it can handle. While more powerful implementations of the method are under development

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[22]The wide range is due to variation in the cost of electricity, which affects life-cycle cost, and thus duct design.

(Tsal, 1990), this tool is not yet ready for commercial use. When it is, it will offer the means to substantially reduce the energy used for air handling in buildings, although its applicability is obviously limited to new construction.

PART LOAD (THROTTLED) CONDITION

The simple analysis above is useful in that provides an order-of-magnitude comparison of the components of air handling energy use, and highlights duct losses as the largest of those components. VAV systems typically operate at flow rates less than that required for full load, as shown in **Figure 1.4**, by throttling the air flow in proportion to the cooling load. As mentioned previously, this throttling occurs at the terminal box dampers, and—for systems using VIV fan control—at the inlet vanes. Since the pressure at the terminal box inlets is generally held constant over the range of flow rates, the effective loss

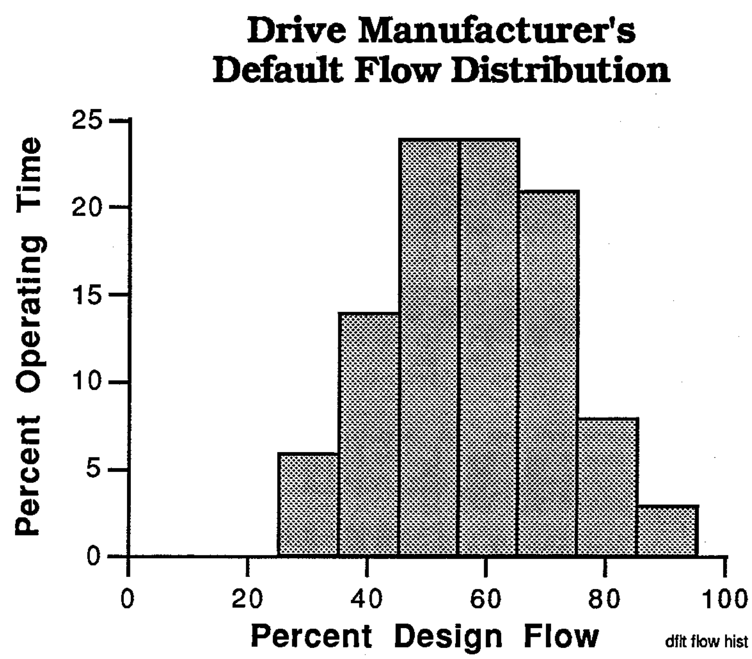

Figure 1.4. Supply fan flow distribution typical for VAV systems. Flow is throttled in proportion to cooling load. The distribution shown is used by one variable speed drive manufacturer to estimate energy savings.

due to this component does not change much with flow. Although duct losses decrease with the square of flow rate, as the vanes at the fan inlet close, the pressure drop across them (in this system) increases to as much as 0.75 kPa (3 in. H_2O). The portion of electrical input energy corresponding to this loss is huge; it is this energy that is saved by using a VSD to control the speed of the fan rather than using VIVs to throttle the flow.

1.2 Saving Fan Energy and Improving Temperature Control in Variable Air Volume Systems

Little work has been done to quantify the savings in fan energy that can be achieved with VSDs, or to identify factors that strongly influence savings. Without comprehensive performance measurements of VSDs in VAV systems, it has been difficult to validate existing methods or develop new methods for predicting savings, and impossible for building designers or managers to make informed equipment selection decisions based on cost-effectiveness criteria. In addition, HVAC control systems employing VSDs are not designed to achieve the higher efficiencies that become possible by incorporating into the control strategy an understanding of the factors influencing efficiency. As direct digital control (DDC) systems become the standard in new construction, the realm of what is possible expands, as all of the components of a building HVAC system can now "talk" to one another. Control techniques for new components such as VSDs and DDC terminal boxes will best evolve if guided by a physical understanding based on actual measurements of component performance.

This motivated the first part of this study, comprising Chapter 2: measurement and analysis of fan motor performance in a commercial office building before and after inlet vanes were replaced with VSDs as a means of controlling supply duct pressure and return air flow. The building used in the study, a 12,000 m^2 (130,000 ft^2) commercial office

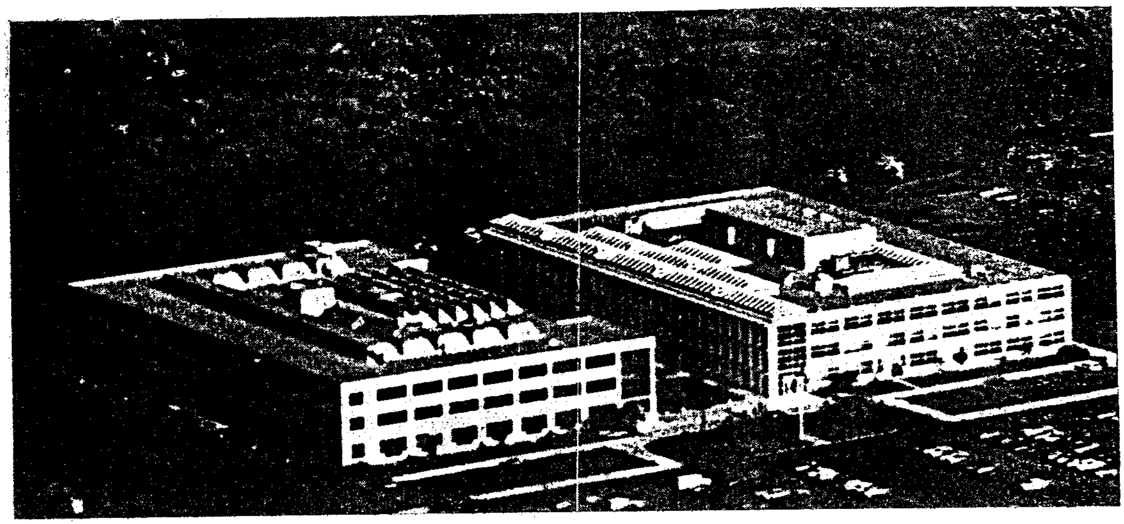

Figure 1.5. Aerial view of the building in which the variable speed drives were installed on fan motors, replacing inefficient variable inlet vanes. The building (on left), located in central New Jersey and constructed in 1983, houses 12,000 m^2 of commercial office space.

building known as Enerplex South,[23] and its twin (**Figures 1.5, 1.6**) are among the best documented structures in the world. The buildings were originally commissioned by the Prudential Insurance Company as a way to explore energy conservation opportunities and better understand energy consumption in other Prudential buildings. Completed in 1983, and heavily instrumented (100 channels), the buildings have been used ever since for energy studies carried out by the Center for Energy and Environmental Studies at Princeton University.

In November of 1987, in an effort to continue the use of Enerplex South as a test bed for ventilation studies, Prudential installed variable speed drives on two supply fan and two return fan motors. The drives, which control airflow by varying the speed of the fan motors, replaced less efficient variable inlet vane (VIV) pressure and flow control, which throttle the flow at the fan inlet. Already instrumented in the course of previous studies of the building (Norford, 1984, and Hsieh, 1988), these fans provided an ideal opportunity to

[23]Recently changed to "2 Research Way."

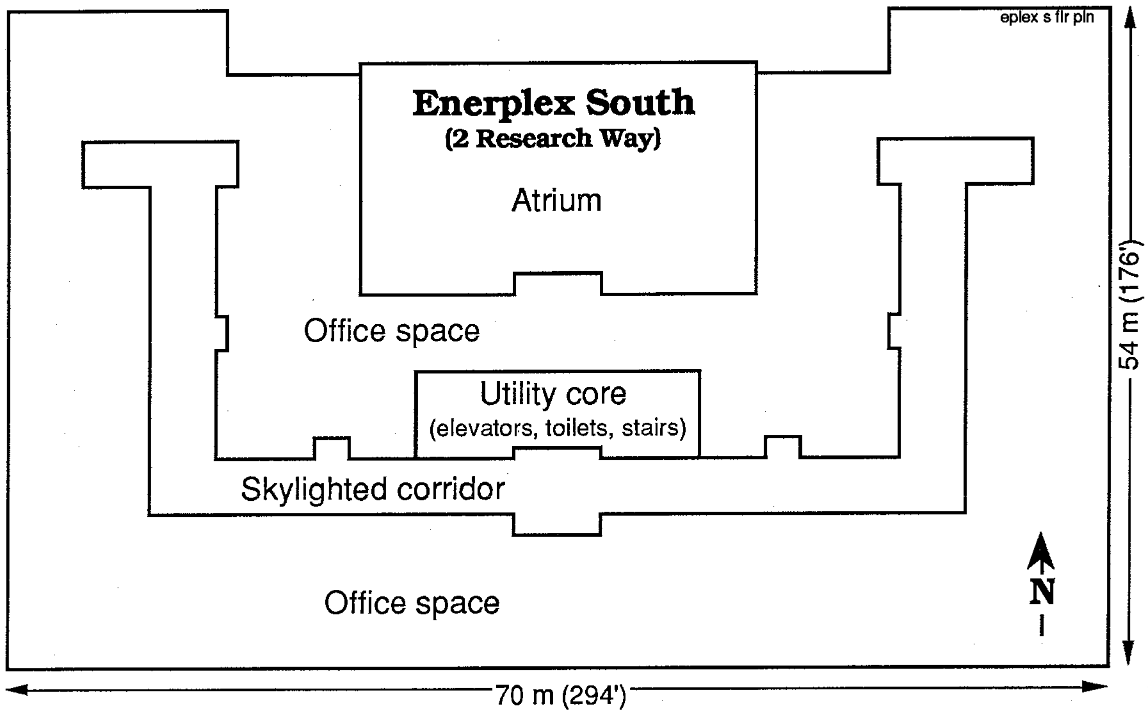

Figure 1.6. Floor plan of the study building (floors one through three similar). The skylighted corridor is open from floor to roof. The atrium, enclosed in glass, is not heated or cooled.

measure and model part-load fan performance and annual energy use for VIV and VSD flow control, estimate annual energy savings due to the retrofit, and investigate the applicability of these results to fan systems in other buildings.

The main control system in the study building is pneumatic, and has no communication with the terminal boxes that would allow the control system to know that a given terminal box is "starved" for air. Instead, the present system maintains duct static pressure at a "design" or "worst-case" level. If duct static pressure were not adequate to meet the flow requirements of a given zone during the cooling season, the space could overheat; in the heating season, ventilation air would fall below the minimum required.[24] The strategy of keeping static pressure high enough for the worst situations uses a maximum amount of fan

[24] With pneumatic terminal boxes, the operator must rely on the "occupant complaint feedback mechanism" to detect such problems.

energy relative to any strategy in which duct static pressure might respond to variable needs.[25]

Our study has shown that energy used by fan motors can be reduced perhaps as much as 50% relative to conventional control (i.e., using inlet vanes and a constant static pressure set point) if, in conjunction with the use of variable speed drives, the constraint on duct static pressure is removed. This is in fact just becoming possible with the introduction of DDC terminal boxes. In a system where terminal box controllers are connected via a network to a central building control system, duct static pressure could be minimized, subject to the constraint that no terminal box is starved. There are no commercially available systems that make use of such a control strategy. Thus, the other major component of this study addresses, in Chapters 3 and 4, how this strategy would be implemented and possible problems that might arise. An instrumented DDC terminal box was installed for experimental purposes in the study building, and various tests were performed to study the behavior of the box, its interaction with the supply fan, and the feasibility of static pressure minimization control.

Experimental evaluation of advanced HVAC control strategies is constrained in a test installation, especially in this case where the main control system is pneumatic. In order to overcome these limitations, a computer simulation tool was employed to construct a numerical model of the terminal box, zone, and supply fan system; physical parameters were based on data recorded during the field experiments. Two static pressure minimization control methods—a modified proportional-integral (PI) algorithm and a heuristic method—are proposed in Chapter 4, and are evaluated using simulation. Both methods use feedback from the terminal box controllers as input to a supply fan controller. We present the results of preliminary parametric studies performed in order to ascertain the influence of the con-

[25]Despite the inefficiency of this, it is easy to see why it is done, if one considers that occupant health and comfort are of much greater concern than energy consumption to building operators. Energy expenses in a commercial building are typically at least an order of magnitude lower than the salaries of its occupants.

troller parameters. Although further exploration is necessary before practical implementation of either of these control methods, the results clearly demonstrate their effectiveness in static pressure reduction, and also elucidate problems associated with parameter selection. Another result of the modeling process was the development of a DDC terminal box model for use with the public domain simulation program *HVACSIM+*, as well as several enhancements to existing models of controllers and other components. The terminal box model was validated using experimental measurements of an installed terminal box.

The operation and maintenance of a VAV system normally involves a procedure called balancing, in which the air distribution system is adjusted to provide the specified flow at all the diffusers. This is done during the startup of a new system and periodically thereafter. During this procedure, the operating parameters of terminal boxes are adjusted. If the specified maximum flow rate set by the balancer is not high enough to handle the cooling requirements of the space (which may change with occupancy and other factors), overheating will occur.[26] Digital terminal boxes have the capability, in theory, of being automatically reconfigured by a central control system if such a condition occurs, although no commercially available fan-powered box does this yet.[27] A more serious problem is a box that is undersized for the load in a space. Both situations are similar, in that the problem box must somehow be identified. We explore some of the ramifications of "adaptive re-balancing," such as selection of a new value for maximum flow.

A problem with terminal box control that is inherent even to properly balanced boxes is steady state error, i.e., the steady state value of room temperature will differ from set point by some offset. This is an artifact of the proportional-only temperature control method embodied in the mechanism of pneumatic box controllers, and should not be a limitation of a digital controller, which could easily implement PI control.[28] Although some

[26] Likewise, if it is too high, overcooling may occur, but this is less likely, since the box can throttle down to the minimum flow rate.
[27] One manufacturer will soon include the capability to manually re-balance a terminal box remotely.
[28] Pneumatic PI controllers are available; their high cost, however, prohibits their use on terminal boxes.

digital terminal box controllers use a PI algorithm to control room temperature,[29] others (such as the one instrumented in this study) use algorithms designed to mimic their pneumatic predecessors—manufacturers have not, as a whole, endorsed PI room temperature control. Whether or not a given terminal box controller currently uses PI room temperature control is of little consequence, as an algorithm could be implemented in existing (installed) DDC terminal boxes through a simple change of "firmware" (software on a chip). In this study we do not attempt to analyze the benefits or problems of PI room temperature control. We do, however, touch on some of the associated issues leading to its ambiguous position in the marketplace.

In addition to evaluating the general usefulness of simulation tools such as *HVACSIM+* for VAV control design, some of the issues associated with real-time use of simulation for improved energy efficiency are explored in Chapter 5.

VARIABLE AIR VOLUME SYSTEMS

VAV system designs take many forms (Rickelton and Becker, 1972). For the sake of introduction, however, the form presented here is the one dealt with in this study—variable system flow rate, both variable and constant zone flow rates, and reheat at the individual zones. At the risk of being redundant, this type of system will be described here in more detail in order to prepare the reader for the material in Chapter 2.

A supply fan supplies "primary" air to all the zones, at a relatively constant temperature year-round. Terminal boxes control the flow of this air so as to provide ventilation and regulate zone temperatures according to local set points.[30] Air is removed from the zones by a return fan, mixed with fresh air and then cooled if necessary before reentering the supply fan. The flow through the return fan is often controlled so as to be less than that

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[29]All DDC terminal box controllers some form of PI control or equivalent for regulating supply air flow.
[30]Ideally, each terminal box serves a single zone and each zone is served by a single terminal box, but this is not often seen in practice, as architectural design or renovation of interior spaces is done independent of mechanical design.

through the supply fan by the volume specified for ventilation—typically 0.009 m^3/s (20 CFM) per occupant in office buildings[31]—and to provide a building pressure that is slightly positive relative to the outside so that exhaust systems (restroom and other) function properly. Mixing dampers control the proportion of fresh to recirculated air. A widely used control feature called an "economizer cycle" can reduce chiller energy use by increasing the proportion of fresh air during the cooling season if outdoor temperatures are low (<13 °C or 55 °F).

Terminal boxes are of both the constant volume type and variable volume type:

Constant volume terminal boxes. Boxes of this type employ a fan to supply a constant "volume" (more precisely, constant volumetric flow rate) of air to the zone. Primary air is mixed with an induced flow of air from the return plenum, and the temperature of the outlet air is varied by changing the proportion of primary air with a damper at the inlet of the box. This type of terminal box is used in spaces where a constant volume of air must be provided in order to maintain even circulation of room air over a range of ventilation levels, or to lessen acoustic disturbances due to changing flow rates. In addition, fan powered boxes are thought to reduce heating energy use, as they enable local recirculation of air warmed, in part, by internal gains (lights, equipment, occupants).

Variable air volume boxes. Used in applications where uniform air circulation is not required or is provided naturally, this simpler (and less expensive) form of terminal box varies the flow of supply air to the zone by controlling a damper.[32]

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[31]This is the level recommended in ASHRAE Standard 62-1989.

[32]The fan powered boxes described here are often called "series" terminal boxes, as the fan is in series with the primary air stream, to distinguish them from another type of fan powered box referred to as "parallel." Fans in parallel boxes are in series with the plenum air inlet, and operate only under heating or low cooling loads. Parallel boxes, therefore, provide constant air volume only when the fans operate. This type of terminal box is not dealt with in this study.

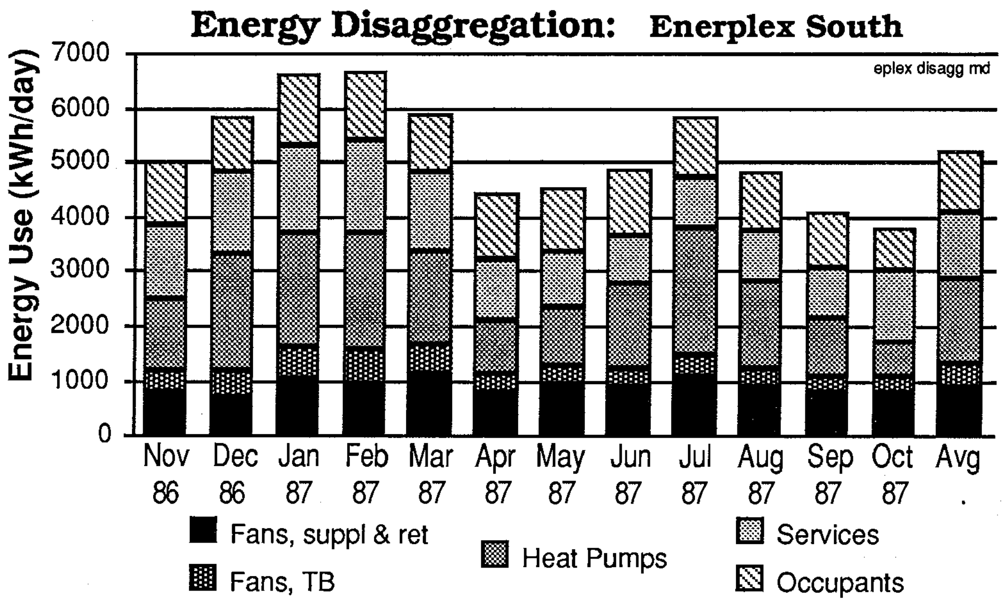

***Figure 1.7.** Monthly and average disaggregation of energy use in the study building, a 12,000 m^2 commercial office building in central New Jersey. Fans are broken down into central supply and return fans, and terminal box fans. "Heat pumps" includes heat pumps and water pumps. "Services" includes lighting of common areas, both interior and exterior; office lighting and equipment is included in the "Occupants" category.*

Both types of boxes may employ "reheat," i.e., during the heating season, heat is added to the air at the outlet or downstream of the box. Reheat is generally provided only in perimeter zones, where heating loads are higher.[33]

A monthly and annual average disaggregation of energy use in the study building is shown in **Figure 1.7**, with terminal box fans as a separate category. While these fan motors are small (fractional horsepower), they are many, and the energy they use should not be ignored:[34] for this period (pre-VSD), the terminal box fans used almost half of the energy used by the supply and return fans.[35] On an instantaneous basis, the small fans are estimated[36] to use a total of 41 kW.

[33]Some inefficient systems in buildings with diverse heating and cooling loads, combined with high ventilation air requirements, use reheat coils year-round to provide heating to some rooms or zones while others are cooled.
[34]Having said this, the author will proceed to ignore energy used by terminal box fans, as it is outside the realm and purpose of this study.
[35]The energy used by terminal box fans varies from month to month for two reasons: sometimes boxes in unoccupied areas are shut off, and fans in perimeter areas run intermittently at night during winter.
[36]Based on rated power and measurements of electrical power made on boxes in Enerplex North.

The fan and supply ducts are sized so as to provide a static pressure at the inlet of each terminal box sufficient to sustain a design maximum flow rate (Walker, 1984). As the air supplied to the zones is throttled by individual terminal boxes, the flow through the supply fan is adjusted to equal the sum of the flow through the zones, by regulating duct static pressure, measured at some point in the supply duct, usually far from the fan.[37] This is usually accomplished by throttling the flow with inlet vanes—dampers at the fan inlet—or by varying the speed of the fan. In a pneumatic system, a proportional plus integral (PI) controller performs this function, with the static pressure signal and set point signal as inputs. In a microprocessor-based digital control system, the function of the PI controller is performed in software. In order to control the return flow rate in a pneumatic system, signals from dynamic pressure sensors downstream of the return and supply fans are fed through "square root extractors" followed by scaling multipliers.[38] These signals serve as input to a PI controller which adjusts the flow through the return fan (Walker, 1984).

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[37] The sensor is typically located in the longest duct run, 2/3 to 3/4 of the distance from the fan to the end of the duct.

[38] Square root extractors convert a dynamic pressure signal to a velocity signal.

Chapter 2

Analysis of a Variable Speed Drive Retrofit[1]

2.1 SYSTEM DESCRIPTION

There are two independent but identical VAV systems in the study building, each serving one side (**Figure 2.1**). Supply and return fans are of the backward-inclined centrifugal type. The terminal boxes are of four types: variable volume (without fans, as described in Section 1.2), with and without reheat coils, which are used in corridors and supply only primary air (from the central supply fan); and fan-powered constant volume with and without reheat coils, in which the fan serves to induce circulation of secondary, or room air. Three-fourths of the boxes are fan-powered, and the majority (located in perimeter spaces) have reheat coils.

[1]This chapter includes work previously published jointly by the author and Norford (1988).

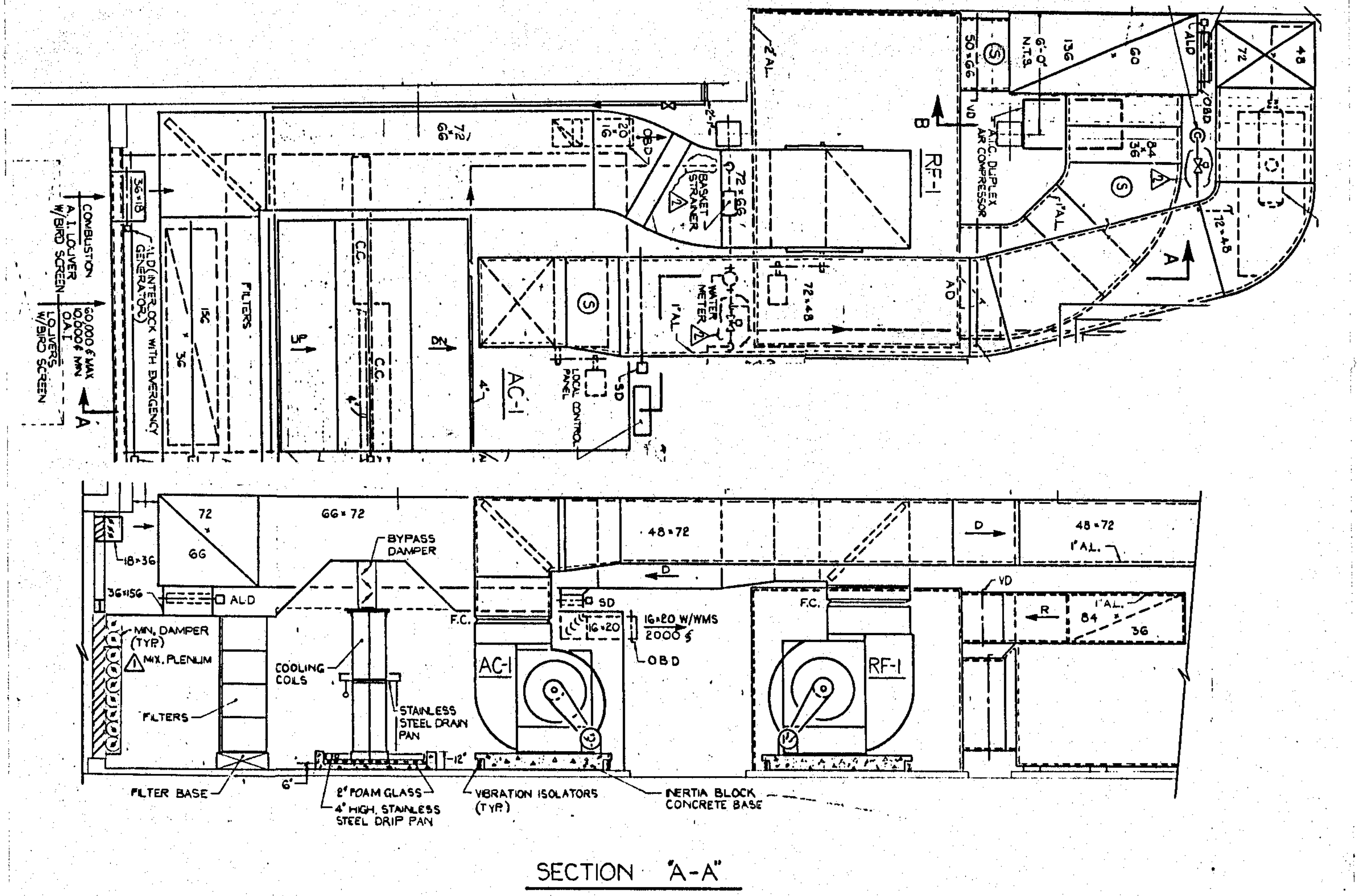

Figure 2.1. Plan and section of basement mechanical room, showing supply and return fan system. This pair of fans serves the west half of the building. The supply fan is controlled to regulate static pressure measured at a sensor in a third floor supply duct. The return fan is controlled to regulate flow a fixed amount below the supply air flow,enabling fresh air to enter the louvers at the mixing plenum. The fresh and return air mix, then pass through filters and cooling coils into the supply fan and to the zones. Source: Flack & Kurtz, Prudential at Princeton: Energy Project, drawing M1-S, revised 4/14/82.

Each box is connected to a local thermostat, and maintains the space temperature by modulating its primary air damper, and in the heating season, modulating the output of its heating coil[2] if the damper is already at a minimum position (**Figure 2.2**).[3] The amount of cooling that a terminal box provides is proportional to the product of the primary air flow rate and difference in temperature between primary and room air. Since supply air temperature is normally relatively constant, cooling capacity is proportional to the primary air flow rate.[4] This flow rate is some function of damper position and inlet pressure. Under normal conditions[5] when duct static pressure (p_s) is regulated by the supply fan at a constant set point adjusted manually by the building operator, the cooling capacity of a box is a function of damper position alone, which the controls at the box vary to regulate primary air flow in response to a changing cooling load in the space served.

As an example, consider a terminal box that is sized and balanced such that it can regulate zone temperature at set point given a cooling load that is equal to or less than some design value. Under this design cooling load there is some inlet pressure at which the box damper would be wide open while maintaining the desired primary flow rate. If the load were to increase without a corresponding increase in inlet pressure, the box would be "starved for air," and the space temperature would increase. Hence, the supply fans must keep duct static pressure at a level such that at any given time, the box with the highest load relative to its capacity is not starved for air. With the existing controls in the study building, duct static pressure set point is set manually and cannot vary automatically in response

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[2]All of the boxes in this building use hot water coils, where a pneumatic valve controls flow.
[3]The details of terminal box operation are covered in Chapter 3; nevertheless, a brief description of the fundamentals is given here in order to build an understanding of VAV system operation.
[4]Cooling capacity here means the rate at which the terminal box can change zone temperature. This rate is proportional to the temperature of air leaving the box, which is proportional to primary air flow rate. Enthalpy and air movement considerations, ignored here, influence the effective cooling capacity, as perceived by the occupants.
[5]i.e., when the required total supply flow is less than or equal to the maximum that can be delivered by the fan without a resulting decrease in static pressure.

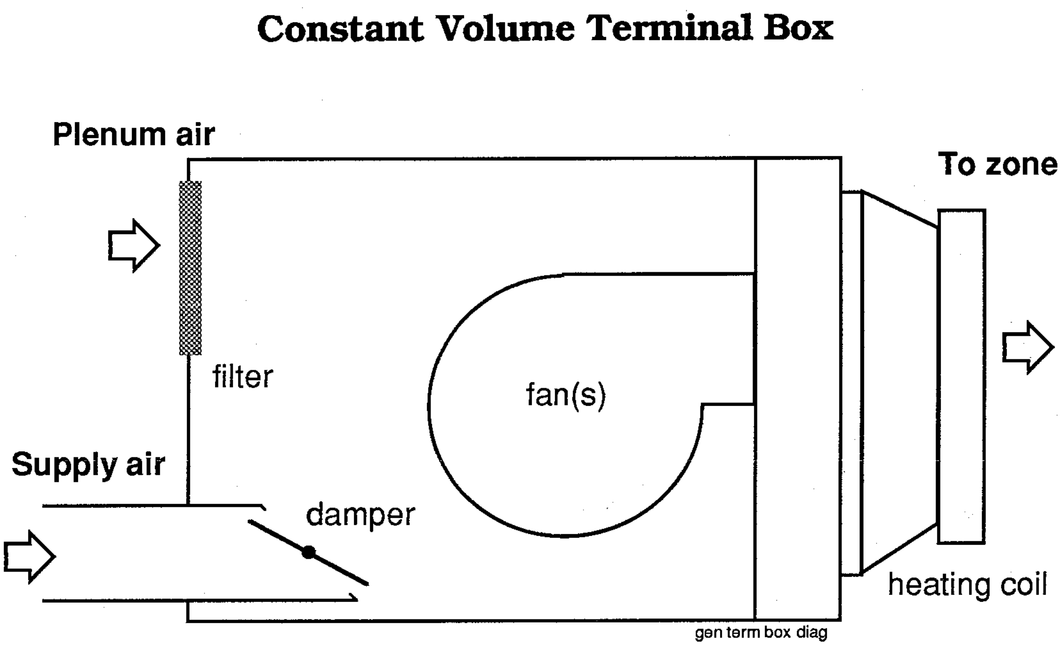

Figure 2.2. Schematic diagram of terminal box. The controls mix constant temperature supply (primary) air with recirculated (secondary) air from the ceiling plenum to vary outlet temperature as required for zone temperature control. Boxes in interior zones do not have heating coils; boxes in corridors do not have fans. Damper shown has a 45° range; pneumatic terminal boxes in the study building have 90° dampers.

to changing loads; it is fixed such that terminal boxes can always handle peak loads in the offices, which for perimeter spaces occur during the hottest months of the cooling season[6]. This static pressure is, for much of the time, considerably higher than the minimum pressure that leaves no box starved. The set point in the study building could be adjusted by the building operator, but in practice has been left at a fixed value.[7]

The VAV control system, including the individual zone control loops, is completely pneumatic. The control system is designed to regulate supply duct static pressure about a

[6]Terminal boxes can also be starved for air in milder weather if the chillers are not operating and the supply air temperature is too high, or if the office space has large internal loads.
[7]Adjusting the set point is not a trivial procedure; there is no direct indication on the control panel of what the set point is—the operator must turn a dial until the actual static pressure comes to the desired value. If all terminal boxes are in a fully open condition (e.g., under very high cooling loads), the maximum static pressure achievable will be relatively low, and it will not be possible to accurately adjust the set point to a pressure above this point. There is, nevertheless, a gauge that displays the controller output pressure, which has been observed to be roughly proportional to duct static pressure, under "normal" flow conditions.

set point—as measured by a sensor located in the duct—and maintain return air flow at 4.7 m^3/s (10,000 cfm) less than supply flow, the required outside air flow rate.[8] Supply pressure and return air flow were originally controlled by VIVs—vanes at the fan inlet which are modulated between a minimum position and fully open. The VIV control was replaced, as mentioned above, and now VSDs regulate static pressure and flow by adjusting the fan motor speed. There is no longer a pressure drop across the inlet vanes,[9] and the fans do less work for a given flow rate.

2.2 VARIABLE SPEED DRIVES[10]

As in most motor applications, maximization of performance and energy efficiency in fan and pump motors can be achieved by properly matching the speed of the motor to varying load requirements. Dampers or valves (such as VIVs), conventionally used to control flow, are inexpensive to install but waste energy. The shaft power to move a fluid varies roughly with the cube of the flow.[11] Adjusting the speed of a fan or pump can reduce energy requirements by as much as 30 to 50%.

There exist a broad range of technologies for controlling motor speed; **Table 2.1** summarizes the major classes. Electronic VSDs, applied to the motor input wiring, have the advantage of being more efficient than shaft-applied drives, and can be installed on existing motors. Recent advances in microelectronics have increased the compactness, efficiency, and reliability of electronic VSDs, and reduced their cost, making them more attrac-

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[8]Observations indicate that the return flow does not track supply flow very well.
[9]Actually, the vanes were left in place, but fixed open; the slight pressure drop that remains across the vanes has not yet been measured, but in the normal range of flow, is less than 30 Pa (0.12 in. H_2O), 6% of the static pressure difference across the fan, according to manufacturers' literature (The Trane Company, 1975).
[10]Most of the background material in this section is based on work by Greenberg et al. (1988).
[11]This is true for systems with a fixed flow resistance coefficient (or system constant); systems with a fixed pressure and varying system constant exhibit a weaker dependence of shaft power on flow rate.

tive to potential users. Although they currently enjoy only a fraction of their total market potential, electronic VSDs constitute a growing share of the market in residential and commercial buildings.

The sort used in the retrofit under study is an inverter-based type manufactured by Parametrics[12], specifically a voltage source pulse-width modulation inverter (highlighted in **Table 2.1**). Inverters work by converting AC power supply to DC, smoothing the waveform with a filter, then converting it back to AC, producing an adjustable-frequency, adjustable-voltage supply that is applied to the stator windings of the motor (**Figures 2.3** and **2.4**). The applied frequency and voltage must change proportionally to maintain a constant motor flux (Pinnella, 1985). Drive output frequency and motor speed vary in direct proportion with the control input signal. The drive studied here uses gate turn-off thyristors (GTOs)—self-commutating power semiconductor switches—in the inverter stage, although newer versions of this model employ Darlington transistors.

Electronic VSDs generally have an adjustable acceleration rate, i.e., the rate at which the output frequency changes in response to a step change in the input control signal. The drive used in the retrofit installation has separate acceleration and deceleration parameters which may be adjusted such that the time required to accelerate the motor from zero to full speed (or vice versa) is from 30 to 300 s (Parametrics, 1985). The control input comes from the output of a pneumatic PI controller which regulates duct static pressure about a set point.

[12]Now ASEA Brown Boveri.

Table 2.1. ADJUSTABLE-SPEED MOTOR DRIVE TECHNOLOGIES

Technology			Applicability (R=Retrofit; N=New)	Cost	Comments
Motors	Multispeed (incl. PAM[a]) Motors		fractional-500 hp R,N	1.5 to 2 times the price of single-speed motors	Larger & less efficient than 1-speed motors. PAM more promising than multi-winding. Limited number of available speeds.
	Direct-Current Motors		fractional-10,000 hp N	higher than AC induction motors	Easy speed control. More maintenance required.
Shaft-Applied Drives (on motor output)	Mechanical	Variable-Ratio Belts	5-125 hp N	$350-$50[b]/hp (for 5-125 hp)	Higher efficiency at part load. 3:1 speed range limitation. Requires good maintenance for long life.
		Friction Dry Disks	up to 5 hp N	$500-$300/hp	10:1 speed range.
	Eddy-current Drive		fractional-2000+ hp N	$900-$63/hp (for 1 to 150 hp)	Reliable in clean areas. Relatively long life
	Hydraulic Drive		5-10,000 hp N	large variation	5:1 speed range. Low efficiency below 50% speed.
Wiring-Applied Drives	Electronic Adjustable Speed Drives	Voltage-Source Inverter	fractional-1000 hp R,N	$1500-$80/hp (for 1 to 300 hp)	Multi-motor capability. Can generally use existing motor. PWM[c] appears most promising.
		Current-Source Inverter	100-100,000 hp R,N	$200-$30/hp (for 100 to 20,000 hp)	Larger and heavier than VSI. Industrial applications, including large synchronous motors.
		Others	fractional-100,000 hp R,N	large variation	Includes cycloconverters, wound rotor, and variable voltage. Generally for special industrial applications.

[a]Pole Amplitude Modulation.
[b]The prices are listed from low to high to correspond with the power rating, which is listed from low to high. Thus, the lower the power rating, the higher the cost *per horsepower*.
[c]Pulse Width Modulation.
Source: "Technology Assessment: Adjustable-Speed Motors and Motor Drives (Residential and Commercial Sectors)." S. Greenberg et al., 1988. LBL-25080, Lawrence Berkeley Laboratory, Berkeley, CA. Voltage source inverters emphasized here.

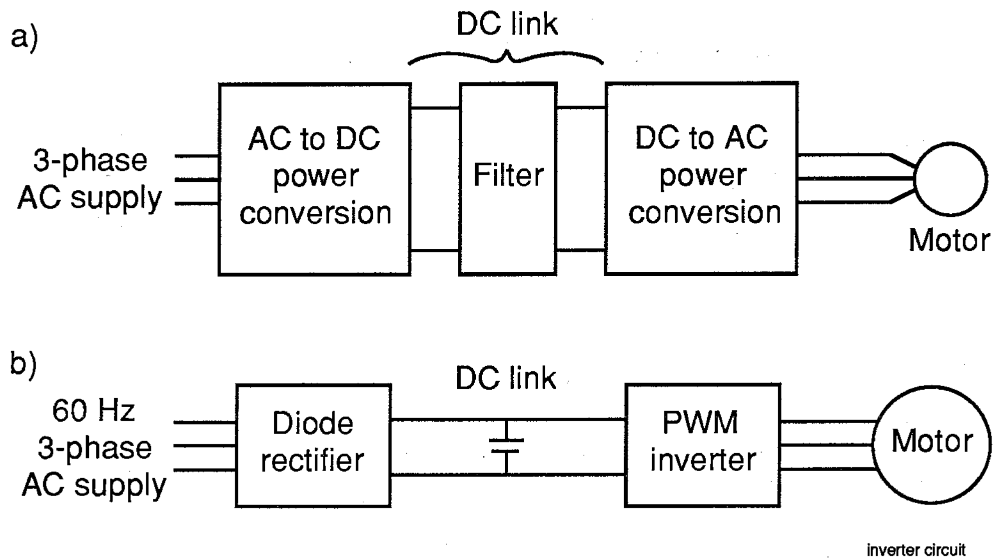

Figure 2.3. a) General inverter power circuit with motor load; b) Basic configuration of pulse-width modulation inverter. Source: "Technology Assessment: Adjustable-Speed Motors and Motor Drives (Residential and Commercial Sectors)." S. Greenberg et al., 1988. LBL-25080, Lawrence Berkeley Laboratory, Berkeley, CA.

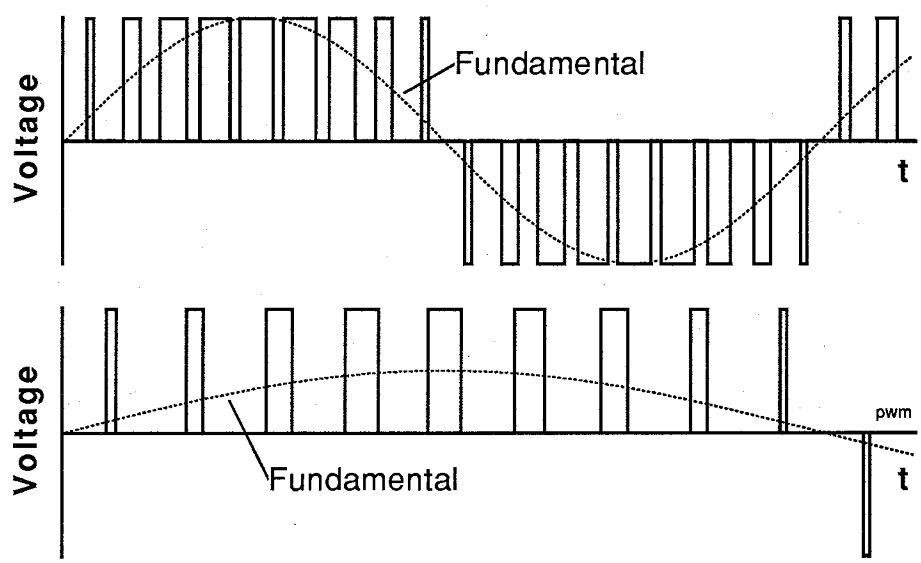

Figure 2.4. Pulse width modulation. Average output voltage, proportional to the fundamental component, is varied by varying the width of constant amplitude voltage pulses. Output frequency is varied by changing the length of the cycle. The output frequency of the second example is half that of the first. In order to maintain a constant motor flux, applied frequency and average voltage must change proportionally; the average voltage indicated by the fundamental in the second graph is therefore half that of the first. Adapted from Jones and Brown (1984).

2.3 EXPERIMENTAL APPROACH

Pre- and post-retrofit annual fan energy use were estimated, and in turn, energy savings. Annual fan energy use for each fan was estimated by combining a polynomial fit of power versus flow data with a flow rate histogram. The power and flow data are hourly averages collected with a data acquisition system for a one year period before the retrofit (the "base year") and for shorter (one to two month) periods afterward[13]. Base year flow histograms were used both in pre- and post-retrofit estimates of energy use so as to "normalize" the energy use, thus removing flow rate distribution as a variable in the comparison of pre- and post-retrofit consumption. Actual metered fan motor electricity use for the base year was used to verify the model.

Since duct static pressure set point is a determinant of fan energy use and is a parameter which is specific to particular systems and control schemes—and one which might be varied in buildings utilizing control systems designed for energy management—we attempted to quantify the effect of changing this parameter on fan energy use in both VIV and VSD systems. Since we did not experiment with changing duct static pressure in the pre-retrofit VIV system of the building under study, we applied results of experiments involving decreasing duct static pressure in a similar building (with identical fans).

The building studied here was modeled in another study (Hsieh, 1988) using DOE-2, a commonly used building energy simulation program. The default power curves DOE-2 uses to model the fan system apparently do not account for regulation of duct static pressure at a constant set point, and result in an under-prediction of annual fan energy use. We

[13]The instrumentation is described in Appendix A.

present the DOE-2 default curves for comparison and quantify the discrepancy in predicted fan energy savings.

2.4 ANALYSIS AND RESULTS

FAN POWER, PRE- AND POST-RETROFIT

The base year for this study was chosen as April 1, 1986 to March 31, 1987. Pre-retrofit estimates of fan power as a function of flow were obtained using base year data. The flow rate data for each fan were reduced to histograms which give the number of hours of operation in the base year as a function of flow rate (**Figure 2.5**). For comparison, a default flow distribution used by one VSD manufacturer is shown in **Figure 1.4.** Regressions were performed to determine expressions for fan power as a function of flow for both supply fans and both return fans:[14]

$$\text{Supply fans: } P_e = 25.0 - 0.36\dot{V}_s + 0.0507\dot{V}_s^2 - 0.000587\dot{V}_s^3 \qquad R^2 = 0.61 \qquad (2.4.1)$$

$$\text{Return fans: } P_e = 9.0 - 0.0888\dot{V}_{ra} + 0.00601\dot{V}_{ra}^2 - 0.000425\dot{V}_{ra}^3 \qquad R^2 = 0.25 \qquad (2.4.2)$$

where P_e is the electric power used at the motor (kW), $\dot{V}$ is the flow (m^3/s), and R^2 is the coefficient of determination. A portion of the data and the regression curves are shown in **Figures 2.6** and **2.7.** The poor R^2 in the case of return fans can be explained by the small slope in the data corresponding to relatively constant power with respect to flow.

Hsieh (1988) compared the regression curve (a) in **Figure 2.6** with the default curve DOE-2 uses in its fan system model (b), which apparently does not account for regulation of static pressure, and found that the latter under-predicts annual fan energy consumption by 14%. Although the DOE-2 Engineers Manual (1982) does not explicitly state

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[14]Fitting both supply fans with one curve was justified by the data (**Figure 2.6**); we know of no physical reason why this should not be the case—the ductwork is nearly identical, with the exception of distribution ducts and diffusers downstream of the terminal boxes, and differences in terminal box adjustment. Data for both return fans were grouped together as well.

that DOE-2 ignores static pressure control, the shape of the DOE-2 power versus flow curve is consistent with those endorsed by ASHRAE (1982, 1988), which correspond to an absence of static pressure control.[15] Static pressure need not be maintained in fixed systems such as boiler exhaust fans, or the return fans used here. Duct static pressure regulation is required, however, for VAV supply fan systems; the set point in the study building is 0.62 kPa (2.5 in. H_2O).[16]

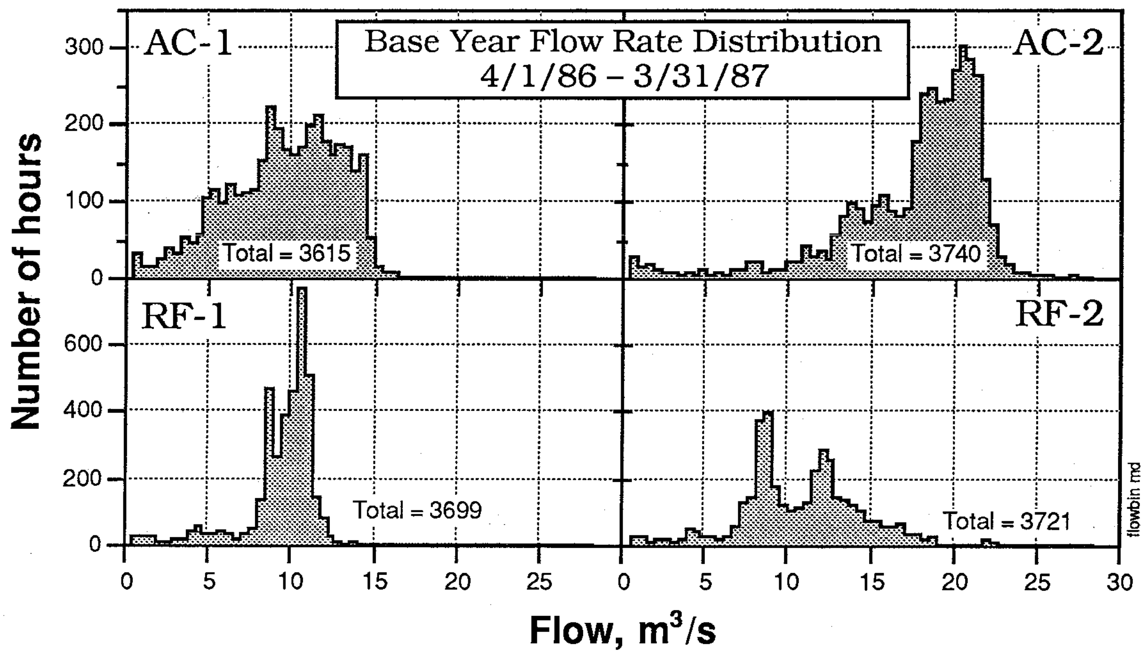

Figure 2.5. Histograms of air flow rates for supply fans AC-1 (west) and AC-2 (east), and return fans RF-1 and RF-2. Data used are hourly samples taken during the base year. The hours were normalized to account for bad or missing data.

[15]These erroneous curves are ubiquitous in the literature; another recent example is Greenberg et al. (1988). Manufacturers' literature as early as 1975 (and probably earlier) shows system curves that properly account for static pressure regulation in VAV systems (The Trane Company, 1975).

[16]Most values of p_s set point presented in this chapter are given as nominal values, i.e. values that the control panel gauge would indicate, within the accuracy of the measurement and gauge. A correlation of gauge readings with independent measurements at the sensor is shown in **Figure 2.15**. Since the original data were collected, the static pressure set point is 6% lower, about 2.35 in. H_2O at the panel.

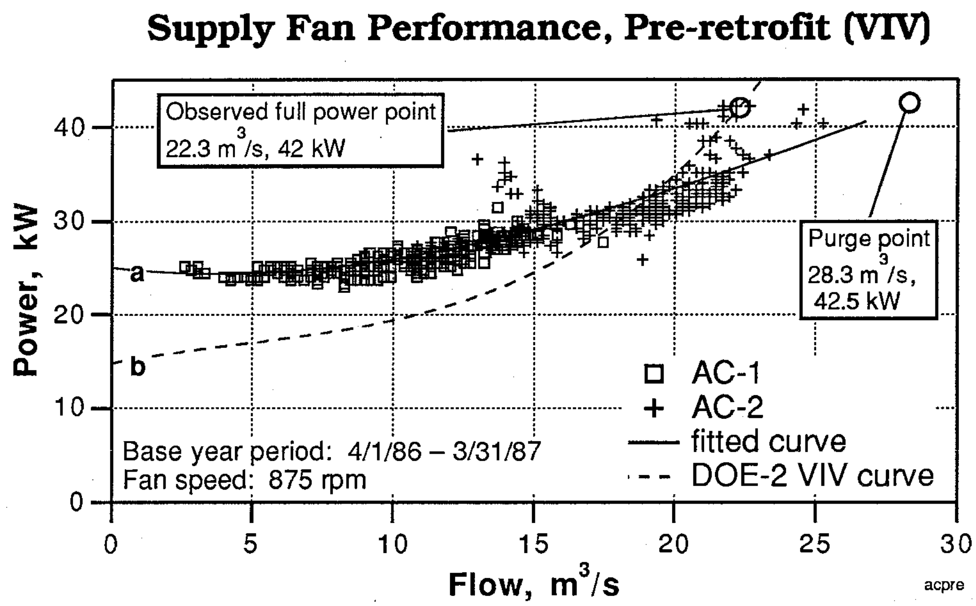

*Figure 2.6. Pre-retrofit supply fan performance, showing a sample of hourly data for the period (curve (a) was fitted using all data in the period). Curve (b) indicates performance predicted by DOE-2, which does not allow for maintenance of duct static pressure The corresponding fan static pressure curves are shown in **Figure 2.12**.*

The DOE-2 curve (b) in **Figure 2.6** originates at the "full power point," 22.2 m^3/s (47,000 cfm) and 42 kW. This point represents the maximum flow and power at which 0.62 kPa (2.5 in. H_2O) duct static pressure can be maintained, obtained from field observations, and confirmed by analysis of the base year data. Nearly all of our data are at flows below this point. The curve is given in dimensionless form by

$$C_p = 0.35 + 0.31C_V - 0.54C_V^2 + 0.87C_V^3 \qquad (2.4.3)$$

where

C_p = power / full power

C_V = flow / flow at full power

Using the measured full flow and power, this becomes

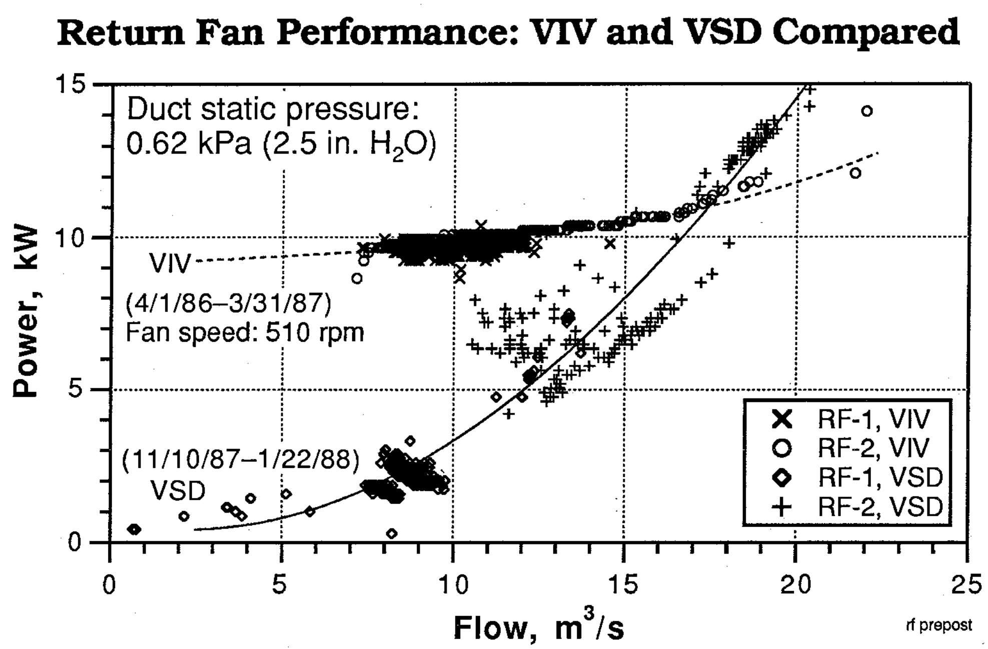

Figure 2.7. Return fan performance before and after installation of variable-speed drives. Points shown for the pre-retrofit period are a sample of the whole set of hourly data.

$$P_e = 14.7 + 0.586\dot{V}_s + 0.0460\dot{V}_s^2 - 0.000334\dot{V}_s^3 \qquad (2.4.4)$$

This curve drops below most of the data (by 0 to 8 kW) because it does not account for static pressure control, although due to the choice of full power point, it over-predicts the power near peak flow for the bulk of the data in this region.

At a flow higher than that at full power is the "purge point," achieved only when all the terminal boxes are wide open (**Figure 2.6**), which we observed when the system was in smoke purge operation.

The HVAC controls literature notes that constant static pressure control will cause deviations from the fan performance assumed by DOE-2 (McQuiston and Parker, 1982 and Wood, 1987); nevertheless, we know of no other field study that has quantified this effect.

Similar analyses are performed on the post-retrofit data, and are shown in **Figures 2.7** and **2.8**. The post-retrofit correlations are

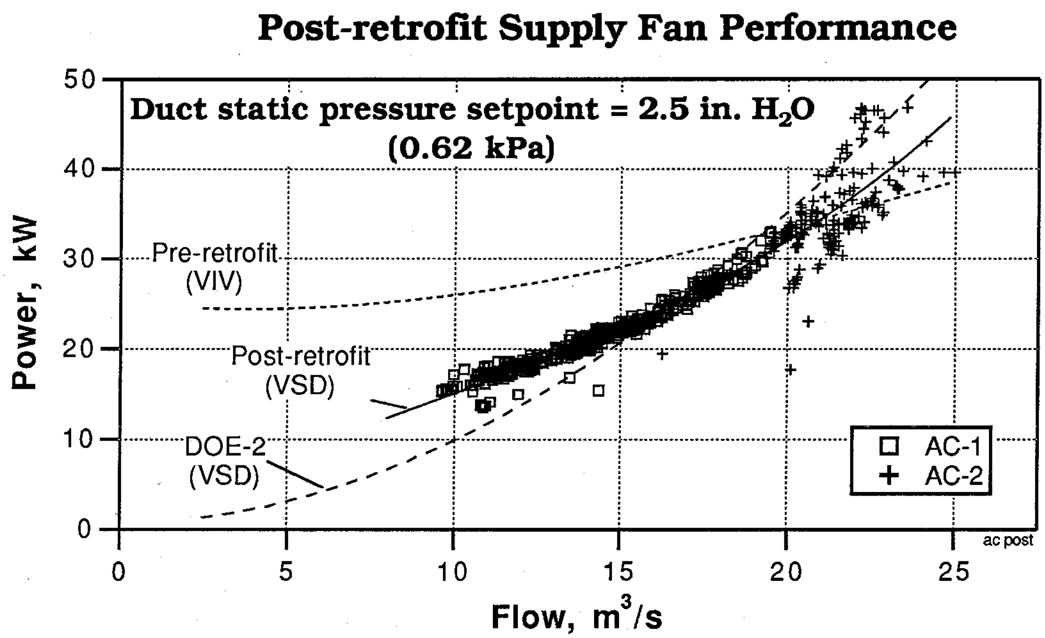

Figure 2.8. Supply fan performance before and after installation of variable speed drives. Pre retrofit curve is a third-degree polynomial fit through a year of hourly data, for the base year period 4/1/86 to 3/31/87. Post-retrofit data are hourly samples for the period 11/10/87 to 12/30/87. The duct pressure set point was 0.62 kPa (2.5 in. H_2O) for data shown. It is notable that the flow rates for fan AC-1 are between 10 and 20 m^3/s for this period, whereas during the base year, flow rates were below 10 m^3/s for about half the operating time, and flows above 15 m^3/s were rare.

Supply fans: $$P_e = -0.91 + 2.1\dot{V}_s - 0.0798\dot{V}_s^2 + 0.00284\dot{V}_s^3 \qquad R^2 = 0.93 \qquad (2.4.5)$$

Return fans: $$P_e = 0.68 - 0.24\dot{V}_{ra} + 0.0533\dot{V}_{ra}^2 - 0.000332\dot{V}_{ra}^3 \qquad R^2 = 0.95 \qquad (2.4.6)$$

The increase in R^2 is due (physically) to less scatter in the data and (statistically) to greater slope. The greater precision of VSD control results in less scatter;[17] the greater decrease in power as flow decreases yields a larger slope. It is easy to see how these characteristics affect R^2 if we express R^2 as

[17]Less scatter may also be due to the fact that the post-retrofit measurement period is shorter and duct static pressure was observed to be fairly constant at 2.5 in. H_2O (gauge reading) during this period. Subsequent observation however, has indicated that p_s occasionally deviates from this value, and the set point appears to drift slightly over time; it is probable that this occurred during the base year. Scatter in these data has also been observed to be related to outside air volume; specifically supply fan power decreases during recirculation mode (no outside air), in part because of higher fan inlet pressures. This may also be due to the lower duct static (and fan outlet) pressures at high flow rates observed during morning warm-up (cool-down).

$$R^2 = 1 - \frac{n \cdot MSE}{\sum (P_i - \overline{P})^2} \qquad (2.4.7)$$

where

n = number of data points

MSE = mean squared error $= \frac{1}{n} \sum r_i^2$

r_i = residual values

P_i = measured power values

$\overline{P}$ = measured mean power

So a decrease in *MSE* and increase in $P_i - \overline{P}$ (greater slope) increase R^2.

The DOE-2 supply fan power curve for VSD control (using the measured full power point) is given by:

$$P_e = 0.643 + 0.00986\dot{V}_s + 0.0945\dot{V}_s^2 - 0.000445\dot{V}_s^3 \qquad (2.4.8)$$

and is shown in **Figure 2.8**. Like the DOE-2 curve for VIVs, it is below the measured curve for a good deal of the flow range, and exceeds it for high flows.

The decrease in power for return fans under VSD control is remarkable (**Figure 2.7**), and helps illustrate how constraining static pressure affects power. Unlike the supply duct system, the return ducts have no moving dampers, so the resistance coefficient or system constant is nearly independent of flow.[18] Duct pressure, and thus fan inlet pressure decrease with flow, and the regression-fit power curve nearly intersects zero power at zero flow, *exactly like the DOE-2 supply fan power curve for VSD control* (**Figure 2.8**).

[18]For a single duct with a single inlet, the system constant would indeed be a constant; branching or multiple inlets, however, cause slight variation.

ANNUAL ENERGY USE AND SAVINGS

A useful basis for comparison of pre- and post-retrofit energy use is a "normalized" annual consumption, calculated using flow distributions observed in the base year. Since the actual flow distributions after the retrofit may vary, this method of normalization does not predict an actual post-retrofit consumption, but serves to evaluate the effect of the motor controls on fan energy use. The normalized annual energy use for a given fan is then

$$E = \sum_{i=1}^{n} H_i \cdot P_e(\dot{V})\Big|_{\dot{V}=\dot{V}_i} \tag{2.4.9}$$

where

n = number of bins in flow rate distribution

H_i = number of run-time hours in bin i for pre-retrofit period

P_e = power as a function of flow determined by regression for given period, evaluated at $\dot{V}_i$

$\dot{V}_i$ = mean flow rate of bin i

As a check on this model, the predicted pre-retrofit consumption for each fan was compared with the actual metered electricity use. The agreement was close, within 3%. Estimated pre- and post-retrofit consumption and energy savings are given in **Table 2.2.** Savings are taken as the estimated base year consumption minus estimated post-retrofit consumption.

The energy-weighted combined savings is 35% of pre-retrofit consumption. Supply fan AC-2 is throttled less than AC-1, and savings are much less (12% vs 45%).Percent savings for return fans are much larger (56% to 68%). The low savings for AC-2 are elucidated in **Figure 2.6**: this fan is operating in the upper range of flow rates most of the time (unlike AC-1), and the motor is usually running near full speed. After this analysis was performed, the building operator discovered several terminal boxes on the east side of the building served by AC-2 that had failed open, but had gone undetected in an unoccupied area of the building. Data from a four month period several months after the repair of the boxes do not indicate any significant change in flow rates. Subsequent to this analysis, at least three more failed boxes were discovered in this half of the building; it is likely that the cause of persisting high flow rates are the result of other such boxes that could easily have gone unnoticed in this less than fully occupied part of the building.

Applying the 35% combined savings to the annual U.S. VAV fan energy use of 14,000 GWh estimated in Chapter 1 gives a potential savings of 4,900 GWh or 16,200 GWh_{th}.[19]

Table 2.2. ESTIMATED FAN ENERGY SAVINGS (MWh/yr)
NORMALIZED BY BASE YEAR FLOW DISTRIBUTION

	Supply fans		Return fans		Total
	AC-1	AC-2	RF-1	RF-2	
Base year (model validation): 4/1/86 – 3/31/87					
Consumption, actual[a]	96.2	118.7	35.4	37.9	288.2
estimated[b]	94.3	118.5	35.8	36.8	285.4
% difference	–2%	0%	1%	–3%	–1%
Post-retrofit, $p_s \approx 2.5$ in. H_2O [c] *11/10/87–12/22/87*					
Consumption, est.[b]	52.0	104.4	11.6	16.1	184.4
Savings, estimated	42.3	14.1	24.1	20.5	101.3
% savings	45%	12%	67%	56%	35%
Post-retrofit, $p_s \approx 1.5$ in. H_2O[d,e] *12/23/87–1/7/88*					
Consumption, est.[b]	28.8	64.5	11.6	16.1	121.0[e]
Savings, estimated	65.5	54.0	24.1	20.5	164.1[e]
% savings	69%	46%	67%	56%	57%[e]

[a]Based on meter readings. Supply fans ran an average of 3,680 h in the base year (= 42% use factor); return fans ran 3,710 h.

[b]Using model combining measured power and flow data with base year flow distribution.

[c]Duct static pressure set point for this period was 2.5 in. H_2O (0.62 kPa), as displayed on the control panel guage. Actual static pressure occasionally deviated from this value.

[d]Duct static pressure set point for this period was 1.5 in. H_2O (0.37 kPa).

[e]Not enough consistent data were available to determine return fan energy consumption at this static pressure; the amounts shown are for return fan consumption for 2.5 in. H_2O.

[19]Assuming 10% transmission loss and 33% mean generating efficiency.

REDUCING STATIC PRESSURE

We recognized that manually reducing the duct static pressure set point from 0.62 kPa to 0.37 kPa (2.5 in. H_2O to 1.5 in. H_2O) would reduce the supply fan power. The return fans are controlled regulate flow rather than static pressure, and to the first order, we expect no change. We chose a set point of 0.37 kPa because this value had been specified at the building design stage and had subsequently been raised by a balancing contractor. According to the manufacturer, the terminal boxes actually require a minimum inlet static pressure in the 0.042 to 0.14 kPa (0.17 to 0.56 in. H_2O) range, the variation being a function of box size, fan speed, an downstream static pressure. The duct static pressure (at the main sensor)[20] required to provide these inlet pressures must be higher than these values, due to frictional and fitting losses between the sensor and the terminal boxes. The value of p_s = 0.37 kPa (1.5 in. H_2O) specified by the building designer was apparently adequate (on paper) for the design flow rates, but could not provide the inlet pressures necessary to enable flow rates capable of handling the higher than anticipated space cooling loads in some zones. We originally hypothesized that the higher value (0.62 kPa) is required only in summer to prevent overheating. However, work by Hsieh (1988) shows high flows occurring at times throughout the year, suggesting a year-round need for higher static pressure to cool some offices during periods of high loads.

Consider, for example, an office far from the building perimeter and therefore subject to a nearly constant load, which contains so much electronic equipment that its terminal box is nearly wide open throughout the year at a given static pressure. It would not be possible to reduce building duct static pressure and still cool the high-load office at a reduced duct

[20] In this building, the static pressure is located on the third floor, halfway (from the fan) down the longest duct run.

static pressure without first providing additional or larger terminal boxes, or lowering the supply air temperature. For a more typical zone, however, it may be possible to lower duct static pressure to below 0.25 kPa (1.0 in. H_2O) before starving the box, as shown in **Figure 2.9**. The data displayed were collected in an experiment that involved manually adjusting the supply fan duct static pressure set point and observing the effect on damper position for one box. In a system with a mix of constant volume (fan-powered) and variable volume boxes, the variable volume boxes are likely to starve first, although in the study building they are located in areas such as corridors, where temperature control is not as critical as in office spaces.

Although only a week of data at a set point of 0.37 kPa was collected at the time of

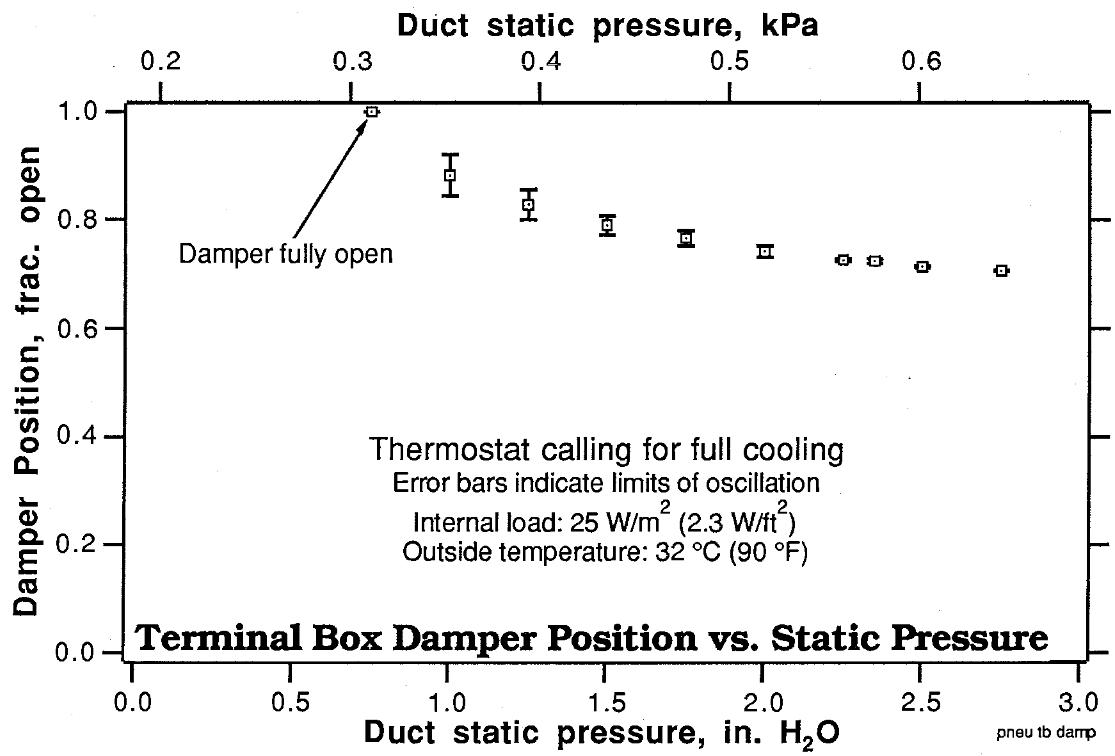

Figure 2.9. Measurements of damper position (fraction of 90°) for a pneumatic terminal box on the first floor of the study building, and duct static pressure at the main sensor. The box has sufficient inlet pressure for values of p_s down to 0.32 kPa, however the office (a southeast corner) was overheating, indicating that the maximum flow limit on the terminal box was probably set too low for the given load conditions.

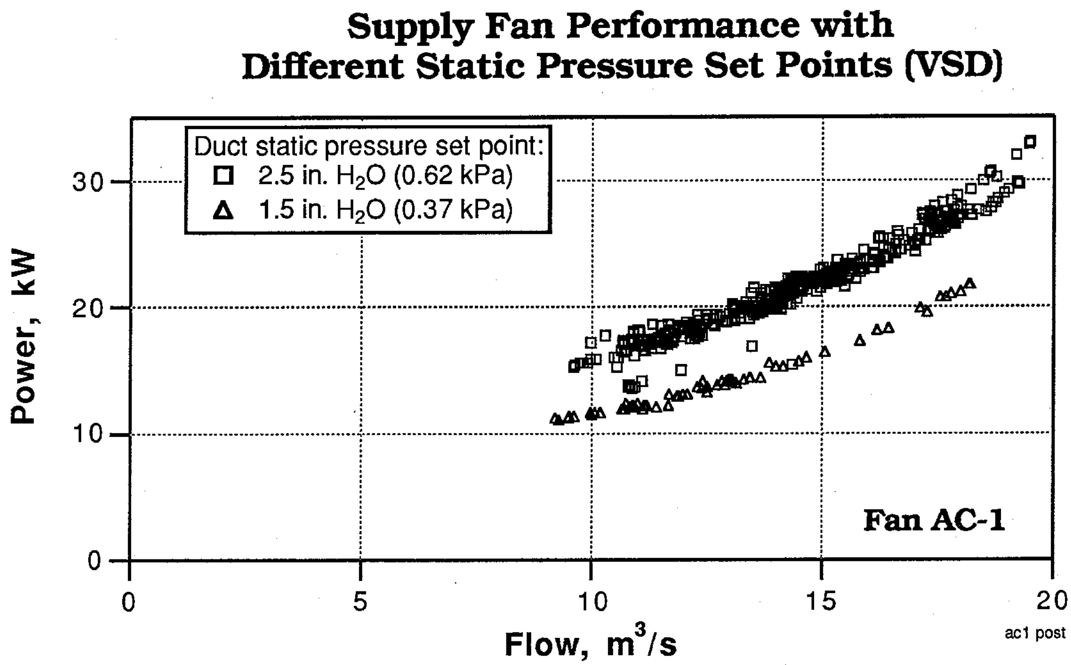

Figure 2.10. Comparison of post-retrofit fan performance showing the effect of changing the duct static pressure set point from 0.62 kPa (2.5 in. H_2O) to 0.37 kPa (1.5 in. H_2O). Data for the 0.62 kPa set point are hourly samples from the period 11/10/87 to 12/22/87; data for the 0.32 kPa set point are from the period 12/23/87 to 12/30/87.

analysis and not enough consistent data could be obtained for the return fans, good results were achieved for the supply fans, and are shown in **Figure 2.10** (for AC-1) and tabulated in **Table 2.2**. The annual consumption reflects operation at a set point of 0.37 kPa for the whole year. This is not realistic, as discussed below, but provides an indication of the influence lowering static pressure has on savings.

The reduction in static pressure results in a clear power reduction, which seems to be independent of flow rate,[21] yielding energy savings over the whole range of flow conditions. The savings jump to 69% for AC-1 and to 46% for AC-2, bringing the energy-weighted average to 57%, not accounting for any change in return fan performance. The increase in savings due to less required fan power is the result of the decrease in pressure

[21]The independence of flow is physically justifiable, as will become evident below.

drop across terminal boxes, which are open wider for a given flow rate. Of course, the set point of 0.37 kPa is somewhat arbitrary; the requirement at any given time may be higher or lower than this. However, if an operating strategy were adopted such that the static pressure were reset by the building operator on a daily basis, this value is probably the lowest set point that could safely be used at this resetting interval.[22]

In calculating the savings at the 0.37 kPa set point, we have ignored an effect of the mechanical operation of the terminal boxes that probably results in a decrease in flow at the fan if static pressure is reduced during the heating season (when our test was done): Once a terminal box damper is at its minimum position, decreasing its inlet pressure will result in less flow through the box (and sub-minimum ventilation air flow). As discussed below, this would not occur with more sophisticated controls, but here it adds to the uncertainty in our savings calculations for 0.37 kPa.

In order to get a better understanding of why VSDs and decreases in static pressure reduce fan electricity use, it is helpful to consider first the simpler case of return fans. **Figure 2.11** relates pressure to flow and shaft power for the return fans; it is typical of (and is based on) fan performance graphs published by manufacturers (with the exception that it shows fan curves for only two speeds—510 rpm and 800 rpm—rather than eight or so).[23] 510 rpm is the speed at which the return fans ran prior to the installation of VSDs (confirmed by our measurements), and remains the maximum speed. A system curve, representing static pressure rise across the fan, as a function of flow rate, intersects the observed full flow point, and the origin.

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[22]Such a strategy would depend on an "occupant complaint feedback system" to determine whether the set point was too low.
[23]The shape of the fan curves is peculiar to backward-inclined centrifugal fans; the general format is the same for all fans and pumps.

In principle, as the flow requirement changes in response to changing supply air flow, the return fan control system responds by closing inlet vanes (in a VIV system) or by slowing the fan down (in a VSD system), moving the operating point down along the system curve; in the (unrealistic) limit of no flow, the static pressure is the same at the fan inlet[24] as it is throughout the return ducts (zero).[25]

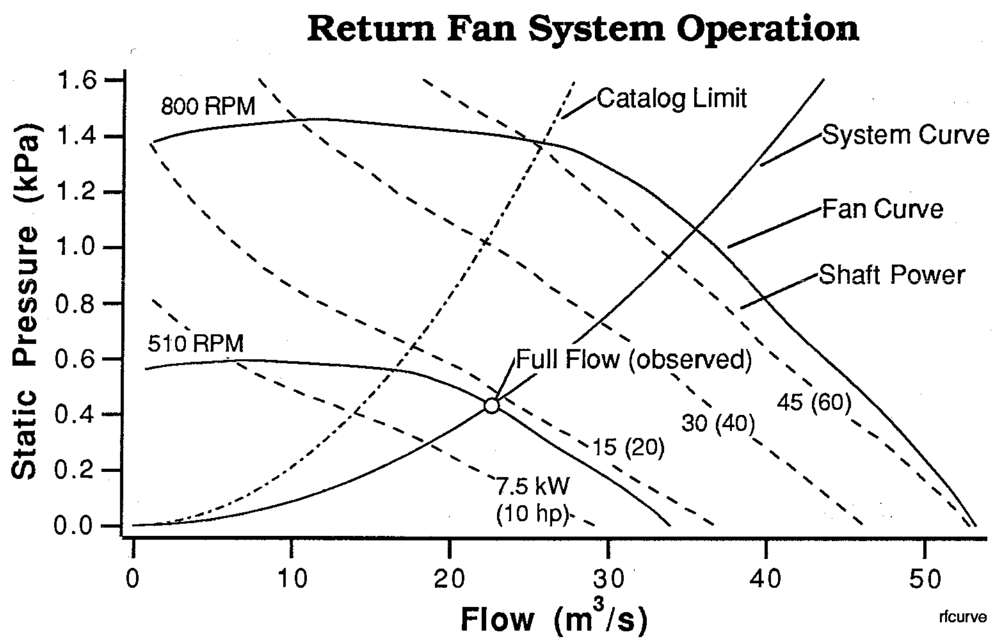

*Figure 2.11. Return fan and system curves for the system in the study building. The corresponding power curves are shown in **Figure 2.7**. The operating point moves along the system curve so as to vary the flow as supply air flow changes. In the return system (unlike the supply system), the ducts have a fixed resistance coefficient; static pressure is not regulated, and therefore drops as flow decreases. Operation to the left of the catalog limit is not recommended, as it can lead to a resonance condition, resulting in vibration. Source of 800 rpm and constant power lines: The Trane Company, fan 49AFDW. 510 rpm fan curve computed from 800 rpm curve using the fan laws.*

[24]Remember that a return duct enters a fan's inlet; a supply duct exits a fan's outlet.
[25]i.e., atmospheric, normalized for pressure differences due to elevation.

Figure 2.12 shows a similar diagram for the supply fans, with a fan curve for 600 rpm and another for 875 rpm, the pre-VSD speed.[26] Here, we have superimposed two VAV system pressure curves, with duct static pressure regulation (a) and without (b). The DOE-2 default model does not include a static pressure constraint, hence its curve (b) intersects the origin; it is identical to a system curve for a duct with a fixed resistance coefficient or system constant, like the return system. Both system curves intersect the 875 rpm fan curve at the observed full flow point. These curves indicate the static pressure rise across the supply fan as a function of flow rate. We expect the system to follow the upper curve (a), intersecting the vertical axis near 0.62 kPa (2.5 in. H_2O).[27] We do not have pressure data either accurate enough or over a broad enough range to confirm this, but the principle is similar to that described for the return fans, the differences being a variable system and constant duct static pressure. As the flow requirement decreases in response to decreasing zone temperatures, terminal box dampers close; the fan control system responds to the increase in duct static pressure by closing inlet vanes (in a VIV system) or slowing the fan down (in a VSD system), in either case moving the operating point down along curve (a); in the (unrealistic) limit of no flow, the static pressure is the same at the fan outlet as at the system sensor (0.62 kPa), and the pressure at the fan inlet equals atmospheric pressure. The VAV system curve (a) is thus the locus of operating points at the intersections of fixed system curves and fan curves—the system curves (which pass through the origin) corresponding to a range of terminal box resistances; the fan curves (similar to those shown) corresponding to a range of fan speeds or (similar but intersecting the pressure axis at a single point, about 5.5 in. H_2O).

[26]The fans were originally specified by the building designer to run at 822 rpm; this was increased at installation.
[27]Actually, 0.69 kPa (2.75 in. H_2O), the measured pressure at this gauge reading.

The "catalog limit" is the fixed system curve that delineates the leftmost limit of system operation recommended by the manufacturer. This particular manufacturer shows this line to the right of the surge zone, by some margin of safety. Operation in the surge zone can lead to instabilities in the fan, accompanied by oscillations in pressure by as much as

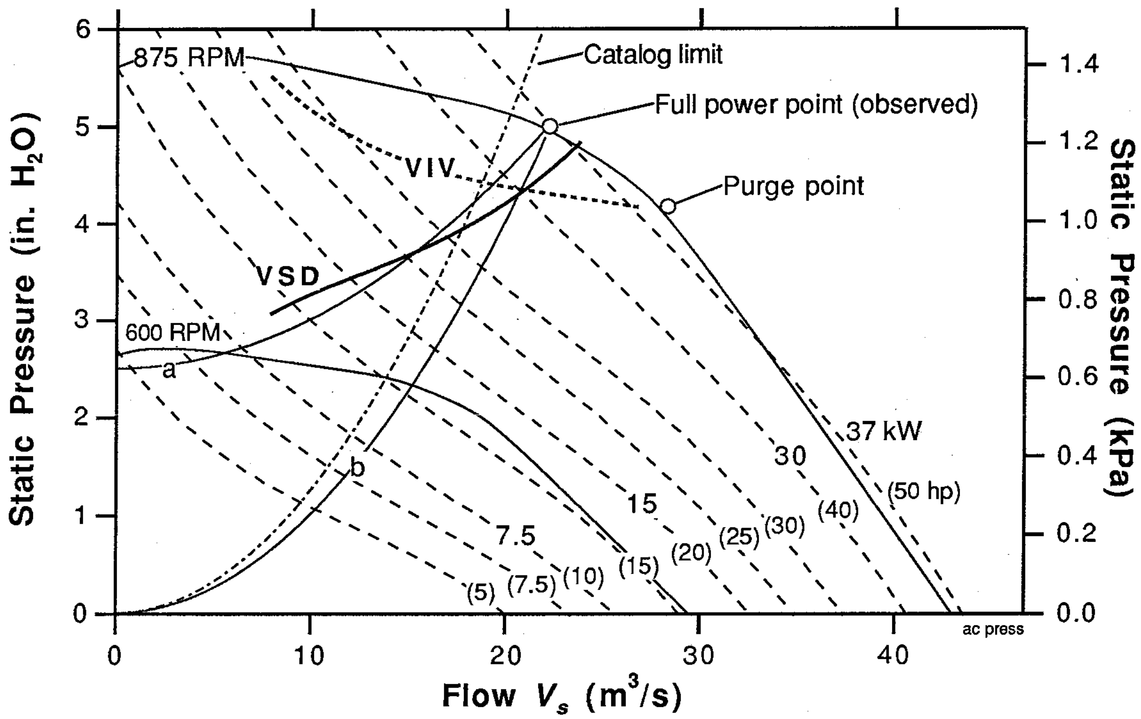

Figure 2.12. Supply fan system curves for the distribution system in the study building (a), which regulates duct static pressure about a fixed set point, and for the DOE-2 default model (b) which represents a fixed system with no regulation of static pressure. The corresponding power curves are shown in ***Figure 2.6****. The VIV and VSD curves are for fan shaft input power calculated from the polynomial fits of pre- and post-retrofit fan electrical power shown in* ***Figures 2.6*** *and* ***2.8****, and an average value of measured motor/transmission efficiency of 0.786. The VSD power curve intersects the full power point on* ***Figure 2.6*** *but not here, because the motor/transmission efficiency measured at this point (and the purge point) is higher than that used to determine VSD shaft power.*

The constant power and speed curves shown are for a fan without inlet vanes. Curves for inlet vanes were not available, however manufacturers' literature (The Trane Company, 1975) gives correction factors as functions of flow; applying this correction to the VIV curve would move it down about 5 to 6% in relation to the power lines.

Note that the original system design specified a design flow of 28.3 m^3/s, at 822 rpm and 1.5 in. H_2O static pressure set point. Raising the static pressure set point brought the system operation to the left of the catalog limit for much of its range, as this fan was not sized for this operating range. Source of 875 rpm and constant power lines: The Trane Company, fan 44AFDW. 600 rpm fan curve computed from 875 rpm curve using the fan laws.

10%, and resulting in a dangerous resonance condition in the fan and duct (The Trane Company, 1982), if not vibration-induced wear on the shaft bearings or scroll (Wise, 1989). Fans are normally sized to enable a range of operation to the right of this line, the lowest flow rate typically one-third of the design maximum (for a"turn-down ratio" of 3 to 1). The original specifications for this supply fan system called for a design maximum of 28.3 m^3/s (60,000 cfm) at 822 rpm and a duct static pressure of 0.37 kPa (1.5 in. H_2O). Given these parameters, a system curve similar to (a) in **Figure 2.12** would have allowed nearly a 3 to 1 turndown ratio (to about 12 m^3/s or 25,000 cfm) outside the catalog limit. Subsequent increase of the static pressure set point and low flow rates (for fan AC-1) has brought system operation to the left of the limit; gross manifestations of surge conditions have not, however, been observed. Operation of the return fans is well within the catalog limit (**Figure 2.11**).

The purge point mentioned earlier is the operating point observed when the system is in smoke purge operation and all terminal dampers are wide open. Under these conditions, we measured a duct static pressure of 0.22 kPa (0.9 in. H_2O). For these supply fans, the increase in duct static pressure from 0.22 kPa to 0.62 kPa occurs at nearly constant fan power, as the flow drops. (This is illustrated in **Figure 2.12**; the path along the 875 rpm fan curve from the purge point to the full power point is nearly parallel to lines of constant power).[28]

The requirement to maintain supply duct static pressure has important energy implications. It may be useful to illustrate the operation of the system in the following manner: Suppose that duct static pressure was controlled by an outlet damper instead of by inlet

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[28]Researchers performing similar analyses should use caution in the choice of the full power point; the full power point as specified in a mechanical system design is intended as a maximum for use in sizing the fans, and is not an operating point at which the static pressure set point can be satisfied. For the building studied here, the full power point as specified in the mechanical system design was close to the purge point, where static pressure is not maintained. In our analysis, we chose the observed full power point.

vanes. The fan shaft power as a function of flow could be determined as the intersection of the 875 rpm fan curve with the constant shaft power curves (**Figure 2.12**). The vertical distance between the fan curve and system curve (a) represents the pressure drop across the damper, and the distance between system curve (a) and the horizontal axis is the pressure drop across the system. The difference in shaft power between the 875 rpm fan curve and system curve (a) is the energy loss due to the damper. VIVs theoretically do better than outlet dampers due to a swirl imparted to the flow (McQuiston and Parker, 1982), and are usually thought to follow a power curve originating at the full-flow point and travelling somewhere between the fan curve and the system curve—but closer to the fan curve—yielding a slight improvement in energy performance.[29] If the static pressure constraint is removed (e.g., in a return system), the pressure drop across the system is less, and the curve (b) intersects constant power lines at a lower level for much of the flow range, corresponding to lower energy use.

A power curve for VSDs, maintaining static pressure, could be expected to track curve (a), as there is no pressure drop due to a damper. A corresponding power curve for a system in which static pressure is not regulated should follow slightly above curve (b) going through the origin. Curves such as (b), which can serve as lower bounds on the energy consumption one could expect from installation of a VSD, are often seen in the literature indiscriminately applied to distribution systems such as this one (i.e., VAV systems), where duct static pressure must be maintained.

The observed power curves, before and after the retrofit, are also shown in **Figure 2.12** ("VIV" and "VSD"). Since fan static pressure rise was not measured with pre- and post-retrofit electrical power and flow, the transformation to the pressure-flow axes is done

[29]The theoretical power curve for a VIV system nearly intersects the fan curve at the full power point—as the pressure drop across fully open vanes is small—and at zero flow, as the pressure drop across the vanes is zero.

by relating fan shaft input power to electrical power as a function of flow.[30] The relationship is

$$P_e = \frac{0.746 P_s}{\eta_{mt}} \tag{2.4.10}$$

where

P_e	=	electrical power measured at the motor (kW)
P_s	=	fan input shaft power (hp), interpolated from manufacturers' curves using measurements of flow rate and pressure rise across the fan
η_{mt}	=	measured motor/transmission efficiency[31] = 0.786 ± 0.003
0.746	=	kW per hp

The combined efficiency term includes losses in the drive, motor, belt, and pulleys[32] (shaft bearing losses are accounted for in fan shaft power curves), and is taken as the mean of 935 measurements at one-minute intervals, displayed in **Figure 2.13**. For these measurements, a sensor was installed to measure static pressure difference across the fan, necessary to determine shaft power.

The VIV and VSD power curves shown in **Figure 2.12** should not be extrapolated below the flow rates for which they are shown, as they are based on an assumption of constant motor and drive efficiency. This assumption does not hold below about 45% of full load, where the efficiency decreases sharply (Baldwin, 1988).

[30]Pressure can then be found as a function of flow and power, determined by fitting a cubic surface to manufacturers' data.

[31]This includes the efficiency of the VSD. The uncertainty given here is the standard deviation of the mean and is deceivingly small; experimental uncertainty is at least one order of magnitude greater. Measurements over a broader range of fan speeds and loads would no doubt be more broadly distributed.

[32]Dividing η_{mt} by the nameplate motor efficiency of 0.902 and the drive efficiency of 0.97 gives a transmission efficiency of 0.90.

The VSD curve is slightly higher than we expect it to be as flow decreases, but not unreasonable considering the scatter in the data, and some allowance for uncertainty in measurement.[33] The VIV power curve is in the general region we expect it to be in; it behaves unexpectedly at lower flows, however, asymptotic to the 25 hp line and appearing to cross the 875 rpm fan curve (were it to continue) which corresponds to the power used by an outlet damper system.[34] If the VIV curve did cross the fan curve at lower flow rates,

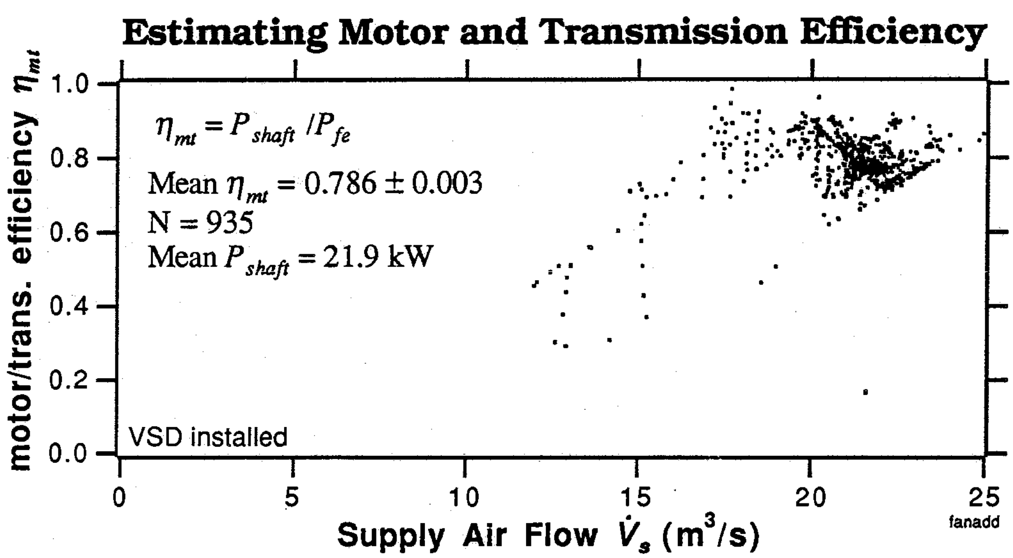

Figure 2.13. Estimate of motor/transmission efficiency (including VSD) for supply fan AC-2 from measured fan electrical power, pressure difference across the fan, and flow. Shaft power is estimated from flow and pressure rise by interpolation between manufacturers' power curves in Figure 2.12. The uncertainty shown is the standard deviation of the mean; experimental error is much higher. A more accurate estimate as a function of load and flow could be obtained through more comprehensive measurements over a broad range of flow rates, static pressures, and fan speeds. This might help explain the seemingly downward trend at lower flows.

[33]The VSD power curve does not go through the observed full power point as it does on **Figure 2.5** because the motor/transmission efficiency used to calculate shaft power for the curve differs from the apparent efficiency at the full power point. More comprehensive measurements of efficiency would be necessary to reconcile this discrepancy.

[34]The VIV curve is plotted very slightly lower than it should be, as the same efficiency—with the VSD in place—was used to calculate shaft power for both curves.

this would indicate that the pressure drop due to the inlet vanes is greater than that for an outlet damper—something not readily justifiable from a physical standpoint and contrary to our expectation that as flow approaches zero, the power lines for VIV and outlet damper should converge on the same point (at about 15 hp).[35] Two factors that could contribute to the discrepancy are

- The calculation of shaft power is faulty, because the estimate of pulley/belt efficiency is inaccurate, it is not (as assumed) constant over the range of operation displayed here, or the actual performance of the fans as installed differs from manufacturers' performance measurements.
- The flow at the inlet is extremely complex, and it is conceivable that the vanes could be increasing the entrance effect, rather than decreasing it.[36] Investigation of such phenomena are, however, beyond the scope of this study.

We have nearly ruled out as a cause for the discrepancy the assumption of constant motor efficiency by applying a typical efficiency curve for this type of motor to the VIV power curve. The constant-efficiency VIV curve shown in **Figure 2.12** comes within a few percent of a variable-efficiency one for the range shown, obviating the use of the latter. An actual efficiency curve from the motor manufacturer would, of course, be necessary to ascertain this.

REDUCING STATIC PRESSURE IN VIV SYSTEMS

In order to gauge the magnitude of savings that could be achieved by lowering static pressure in a VIV system, we performed tests at two set points in a very similar building with identical fans controlled with VIVs.[37] In a VIV system, when the duct static pressure set point is decreased, the inlet vanes close, and terminal boxes open up to maintain the

[35] Buffalo Forge Co. (1983) provides a good illustration of this. Compare Figures 15.23 and 15.25 in this reference.
[36] ASHRAE (1982) warns that improper design of inlet vanes can result in an effect opposite to that intended.
[37] Enerplex North—the other building shown in **Figure 1.5**.

same flow. The pressure drop across the system is less, but this pressure drop is transferred to the inlet vanes, and a reduction in power should be observed, due to the greater swirl that the more highly angled vanes impart to the inlet air. We had hypothesized, based on published power curves for VIV control (Buffalo Forge Co., 1983) that this reduction in power is perhaps about 15%. The data shown in **Figure 2.14** indicate no significant savings resulting from decreasing duct static pressure set point from 0.67 to 0.42 kPa (1.7 to 2.7 in. H_2O)[38] in a VIV system. Our expected savings could, however, be obscured by noise in the data (~15% at 25 kW).

Comparing published power curves for VIV and VSD leads us to expect varying duct static pressure to have a larger impact on VSD than VIV. In fact, we see a 34% (of post-

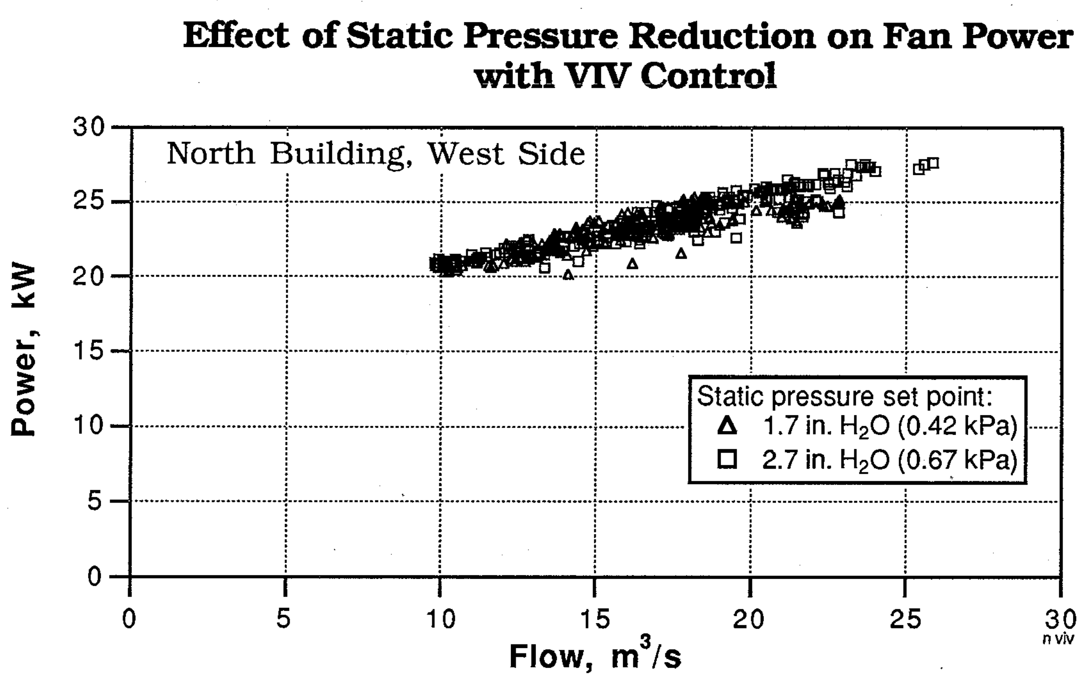

Figure 2.14. Comparison of fan power at two duct static pressure set points in a VIV system. Data are hourly samples from a building similar to the retrofit building (with identical fans), for the period 5/2/88 to 6/11/88.

[38]As measured on the panel gauge.

retrofit annual energy use at 0.62 kPa duct static pressure) reduction for this VSD system.

Fan Performance over a Range of Static Pressures

From the data presented so far, it is difficult to visualize how far power and flow in the VSD system vary over a range of duct static pressures. For this purpose, a sensitive linear pressure-to-voltage transducer was installed to connect the pneumatic control system's static pressure sensor to the data acquisition system; data were gathered at one-minute and five-minute intervals[39] during the summer of 1989, in a series of experiments in which the static pressure set point was adjusted manually. **Figure 2.15** shows the relationship between pressure readings displayed by the control panel gauge and micromanometer readings made at the duct sensor. A second calibration (not shown) of data returned by the transducer to micromanometer measurements was used to obtain accurate recorded values of static pressure. The measurements show that the pressure indicated by the gauge is about 0.25 in. H_2O low at the upper end of the range shown.

Figure 2.16 shows the results of these 2,559 measurements of power, flow, and duct static pressure for fan AC-2.[40] A plot of P_e vs. p_s shows the large energy savings possible due to static pressure set point adjustment, approximately 50% going from 0.62 kPa to 0.25 kPa (1.0 in. H_2O). The scatter in power observed here for a given value of p_s is primarily due to variation in flow rate, corresponding to a range of zone cooling requirements.

These experiments were conducted from late June through early September, for the most part during the day; outdoor temperatures were in the 24 °C to 35 °C (75 °F to 95 °F) range, while loads were light (this half of the building was only 35% occupied at the time).

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[39]One-minute intervals were used during periods of manual static pressure adjustment.
[40]The reader should bear in mind that the flow rates for this fan are normally confined to a narrow range at the high end, due to problems discussed earlier.

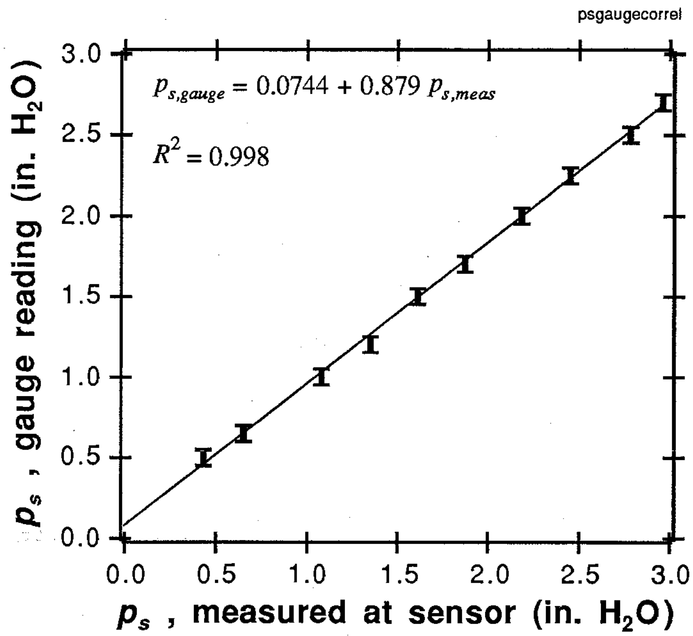


Figure 2.15. *Relating static pressure readings at the fan control panel in the basement mechanical room to measurements made at the duct sensor on the third floor. The pneumatic sensor measurement is converted by a transducer to a control pressure between 0 to 20 psig, connected to the gauge by a pneumatic line. An EDM digital micromanometer was used for manual measurements, which also served as the basis for calibrating automatic measurements made by the data acquisition system, connected to a sensitive electronic transducer on the gauge pressure line.*

Total horizontal solar radiation ranged from 200 W/m² to 800 W/m². Fan power time series segments were smoothed to diminish noise due to low meter resolution.

A number of salient features distinguish the results. Except for a group of points corresponding to $p_s \approx$ 0.62 kPa (2.5 in. H_2O) and $P_e \approx$ 25 kW, fan power is significantly lower (as a function of flow) than that measured for the VSD system under constant static pressure set point of 0.62 kPa, shown in **Figure 2.8.**

Determining a Lower Bound on Static Pressure

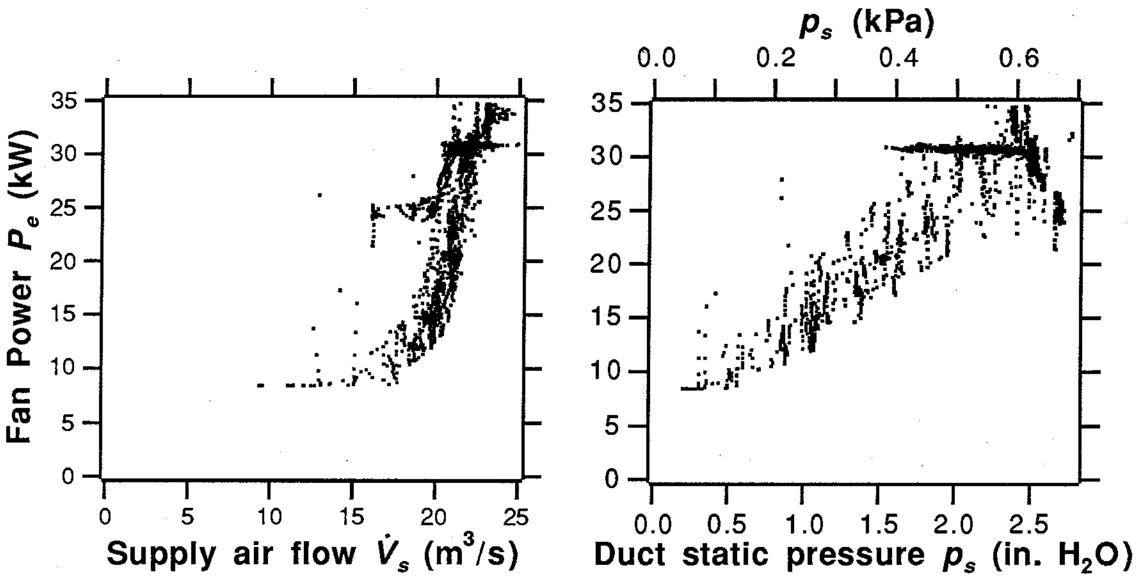
Throughout the experiments, flow was observed to vary somewhat in direct proportion to changes in static pressure; this can be attributed to those terminal boxes that are fully open, or fully closed, and the slight pressure dependence of pneumatically controlled boxes (discussed further in Chapter 3). The response of flow (graph (c) of **Figure 2.16**) is fairly linear down to $p_s \approx$ 0.25 kPa (1.0 in. H_2O), below which the data exhibit a square relationship more typical of a system with a fixed system constant (constant flow resistance coefficient), i.e., $\Delta p = k\dot{m}^2$, where $\dot{m}$ is the mass flow rate. Thus in this region, most (if not all) of the dampers are wide open. In addition, the scatter tapers toward the origin, indicating a uniform response. The variation in flow at higher static pressures is due to differing load conditions and corresponding damper positions. The significance of the system behavior below $p_s \approx 0.25$ kPa is that it defines an upper bound on static pressure reduction for this building under summer conditions.[41]

Anomalous Performance

The dark trace of points with a constant power of about 30 kW represents a three day period at the end of August and beginning of September. The conditions during this period differ from those during the period ending three weeks earlier (when the rest of these data were collected)[42] in two respects: an additional 500 m² (5,000 ft²) of office space (in a south-facing perimeter area) served by this fan had become occupied (an increase of about 35%), and the static pressure set point was left constant at 2.25 in. H_2O (2.5 in. H_2O or 0.62 kPa at the sensor). Daytime temperatures for these three days were from 20 °C to 30 °C (68 °F to 86 °F).

[41]At least one of the 65 terminal boxes could be wide open at some pressure higher than this, with no discernable decrease in total flow rate through the fan; hence, the static pressure reduction strategy involving daily manual adjustment discussed earlier would probably require a set point in the neighborhood of 0.37 kPa (1.5 in. H_2O), allowing for some margin of safety.

[42]Data for the preceding evening are shown as well, but the static pressure set point was not constant.



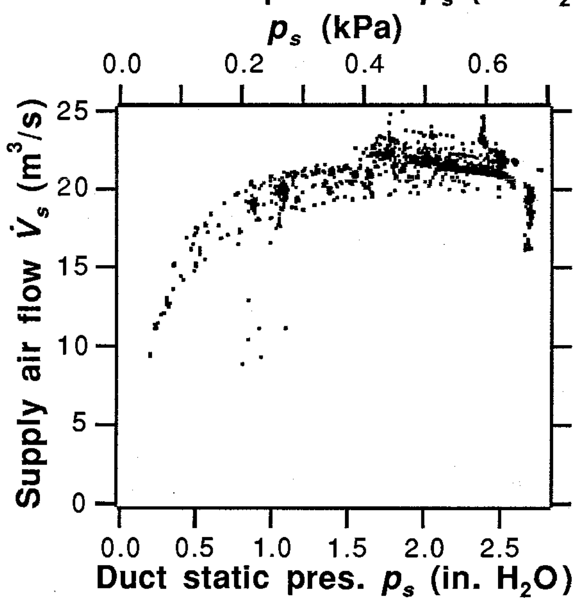


Figure 2.16. *One- and five-minute measurements of fan electrical power, flow rate, and duct static pressure for fan AC-2, made over the summer of 1989. Power has been smoothed (in time series) to lessen noise due to low meter resolution.*

The behavior of the system below 0.25 kPa suggests that this is a lower bound of a feasible static pressure reset range.

The dark trace of points at 30 kW appears to signify a malfunction causing the fan to be unable to reach full speed.

For all three days, system operation follows the same pattern, illustrated in time series in **Figure 2.17**: static pressure is initially between 0.56 kPa and 0.62 kPa (2.25 in. H_2O and 2.5 in. H_2O); flow is around 20 m³/s (42,000 cfm). As the day progresses and loads increase, flow increases to about 23 m³/s (49,000 cfm) by about 2:30 pm, while pressure has decreased to around 0.42 kPa (1.7 in. H_2O); the process then reverses as the loads (to

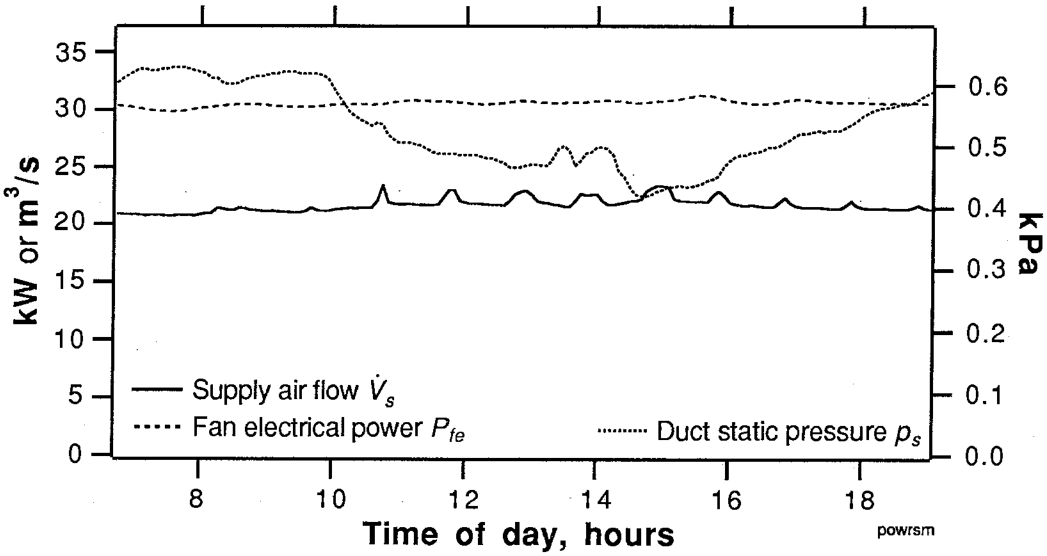


*Figure 2.17. One of the days during which anomalous fan performance occurred, corresponding to the long dark group of points in **Figure 2.16**. The fan cannot maintain static pressure at set point; it appears to be unable to achieve maximum speed.*

a large extent solar) decrease. Power remains nearly constant throughout, since the operating point is moving along a fan curve that is relatively parallel to lines of constant power in this region.

This fan curve, however, appears to be in the 750 rpm region, suggesting that the anomalous behavior is more likely attributable to a fan or power control malfunction than to the increase in conditioned space (i.e., because the speed should have increased to 875 rpm in order to maintain pressure).[43] The scatter on the power vs. flow graph (a) for this period is comparable to that observed earlier for this fan (**Figure 2.5**), suggesting that similar occurrences could be partly to blame.

[43]It is curious, nevertheless, that the west side fan exhibited similar behavior during this period (although static pressure was no quite as low). A similar office space on this side had been finished as well, but was not yet occupied.

Pressure Dependence of Supply Flow Rate

The linear response of flow to changes in pressure above 0.25 kPa can be seen more clearly in **Figure 2.18**, which shows measurements taken over five short (5 min - 1 h) periods during which static pressure set point was varied. The load (and hence required flow rate) during these periods can be assumed to be nearly constant. A regression of data above p_s = 0.25 kPa shows the flow rate to decrease about 7 m^3/s per kPa reduction in static pressure (3,700 cfm per in. H_2O), which would amount to a 2.7 m^3/s (5,600 cfm) variation in flow over a static pressure reset range of 0.37 kPa (1.5 in. H_2O). This is 12% of the pre-retrofit observed full flow rate. Averaged among the 65 terminal boxes served by this fan, it is equivalent to about 0.04 m^3/s (85 cfm) per box.

Although it is impossible to disaggregate this effect into a component related to fully open or closed boxes and one due to pressure dependence of the boxes within their reset range, it can be taken as an upper bound on the degree of pressure independence: pressure dependence on the order of 0.04 m^3/s per box over a 0.37 kPa static pressure range is probably not severe enough to preclude, on this basis alone, static pressure minimization in a pneumatically controlled system.

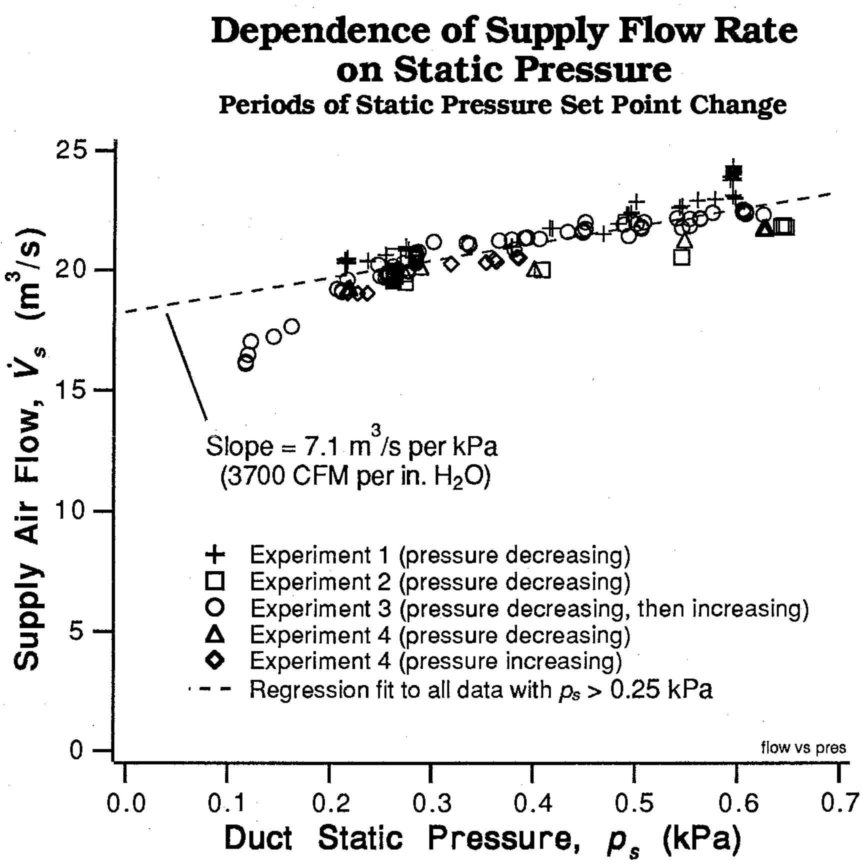


Figure 2.18. Pressure dependence of supply air flow rate on static pressure. Flow and pressure measurements taken on fan AC-2 over five short (5 min - 1 h) periods during which static pressure set point was varied, while the cooling load on the zones was relatively constant.

Comparing Measured Savings with those Predicted Using DOE-2 Fan Curves

Given that the DOE-2 curves do not model constant duct static pressure control, it is useful to compare the savings predicted using DOE-2 curves with those calculated using measured power curves.

The combined savings of 35% is in good agreement with reported results of simulations (which employ the DOE-2 default fan curves) for office building fan systems, which range from 26% to 30% (Eto and de Almeida, 1987). Comparisons between buildings are tenuous, however.

Using power curves presented by Buffalo Forge Co. (1983) as a basis, we expect the savings calculated using the DOE-2 curves for VIV and VSD control to be larger than that of the in-situ results. This is not the case—if we apply the DOE-2 curve to the our histogram of measured flows, we obtain estimated retrofit savings of 30% for supply fans only, compared to 36% for our model. Essentially, this is because the actual VIV energy consumption is greater than that predicted using the DOE-2 curve to a larger extent than the actual VSD consumption exceeds the DOE-2 prediction. Hsieh (1988) did a comparison using DOE-2 simulations of this building, and found that although consumption is underestimated for both VIVs and VSDs, the difference, or savings, is in close agreement with measured results.

2.5 Cost Effectiveness

In light of the additional summer measurements presented above of supply air flow and fan power over a range of duct static pressures, it is clear that the energy savings estimated at a duct static pressure set point of 0.37 kPa (1.5 in. H_2O) is not necessarily an upper limit; the effect of lower pressures on total system flow indicates that it might be possible to go below this set point, although it is impossible to determine without communication with each box. Estimating the energy savings due to a static pressure minimization control scheme using DDC terminal boxes is complicated by the diversity of zone load-to-capacity ratios and box inlet pressures, and ultimately limited by the zone with the worst combination of these at any given time. Simulation can help in estimating savings, as discussed in Chapter 4, but would require incorporating this diversity into the system model, a task that was beyond the scope of this study.

For the purpose of a simple payback evaluation, we assume that the static pressure set point can be reduced to 0.37 kPa for nine months (September to May).[44] Again, this is probably the lowest value that could safely be used if the set point were adjusted manually on a daily basis. As such, the cost of additional automatic controls is not included.[45]

The payback analysis, summarized in **Table 2.3**, is based on the following:

- Comparing peak fan power before and after the VSD retrofit reveals that there is no demand savings when the static pressure is maintained at 0.62 kPa (2.5 in. H_2O). If static pressure is reduced to 0.37 kPa (1.5 in. H_2O), the demand savings (for supply fans only) is about 20 kW.

- Since the demand is billed on a "building heating service" rate, which includes a demand credit and minimum *billed* demand, the avoided billed demand is less than 20 kW. For nine months in the period 3/87 to 2/88, this amount would have averaged 13.4 kW (had the static pressure set point been 0.37 kPa), and that is what is used in this calculation.

- The cost of the avoided demand is \$8.17 (on peak) + \$1.06 (off peak) = \$9.23/kW-month for nine months. The cost of energy is \$0.051/kWh.[46]

At 0.62 kPa duct static pressure, a simple payback period of 7.7 years is too long to make VSDs a good investment. Property managers consider two to three years to be attractive. Even if the flow rates for the east side fan were comparable to those for the west side, the payback would still be too long at 6.1 years. At a reduced duct static pressure of 0.37

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[44]We make simplifying assumption of constant monthly use: the energy savings is simply 9/12 of 164 MWh/yr plus 3/12 of 101 MWh/yr, or 150 MWh/yr, 52% of estimated pre-retrofit annual fan energy use.
[45]Manual checking or adjustment takes only a few minutes and could likely be included in the course of a maintenance person's daily activities with no additional labor cost.
[46]Average rate for consumption only, taken from bills for the period 3/87 through 2/88.

kPa, the savings increases to yield a payback of 4.6 years (or 4.3 years if AC-2 performs as well as AC-1 after all broken terminal boxes are fixed).[47]

If the duct static pressure set point is set manually over a range between 0.37 kPa and 0.62 kPa, depending on load, the savings should fall somewhere in between. If it were controlled automatically by DDC system using feedback from the terminal boxes, savings could exceed the prediction based on maintaining 0.62 kPa throughout the summer, since

Table 2.3. COST, SAVINGS, AND SIMPLE PAYBACK FOR VARIABLE SPEED DRIVE RETROFIT

WITH AND WITHOUT CHANGE IN STATIC PRESSURE SET POINT

	Supply Fans	Return Fans	Total	Total, 1989 prices[e]
Cost[a]	$25,200	$14,800	$40,000	$33,500
Savings, 2.5 in. H_2O				
Energy[b]	$2,900	$2,300	$5,200	$5,200
Demand	—	—	—	—
Total	$2,900	$2,300	$5,200	$5,200
Payback (yrs)	8.7	6.4	7.7	6.4
Savings, 1.5 in. H_2O[c]				
Energy	$5,300	$2,300	$7,600	$7,600
Demand[d]	$1,100	—	$1,100	$1,100
Total	$6,400	$2,300	$8,700	$8,700
Payback (yrs)	3.9	6.4	4.6	3.9

[a]Installed cost in 1987, including design, for drives on two 50 hp and two 20 hp motors (Watson, 1988). Manufacturers' list prices: 50 hp: $8,625; 20 hp: $4,025 (Woodward, 1989).

[b]at average billed rate for consumption only, $0.051/kWh.

[c]Static pressure set point at 1.5 in. H_2O for September through May, 2.5 in. H_2O for June through August.

[d]13.4 kW at $9.23/kW-month for 9 months.

[e]Manufacturers' list prices, 12/89: 50 hp: $6,300; 20 hp: $3,100 (Woodward, 1989). Installed cost = 1987 installed – difference in list price for equipment, i.e., labor is fixed.

[47]These simple payback figures are included for comparison purposes only; a realistic analysis of automatic static pressure control must take into account the higher cost of the requisite digital control, relative to pneumatic control.

peak loads are not seen during much of the cooling season (although the demand portion of the savings would probably not increase). The benefits of such a scheme, of course, would be influenced by occupancy factors that affect internal loads.

The average installed cost of $290/hp is lower than typical for equipment and installation reported in a market survey dated the same year the drives were installed (EPRI, 1987).[48] In the two years since the installation was made, the cost of the drives used has come down markedly (**Table 2.3**): manufacturers' list price for the equivalent 50 hp model as of December, 1989 is $6,300, a decrease of 27% from the 1987 list price of $8,625 (Woodward, 1989). Payback periods calculated with the resulting 16% reduction in cost in installed cost are considerably more attractive, at 6.4 years for 0.62 kPa, and 3.9 years for 0.37 kPa.

In applying these results to other buildings, a major factor influencing payback is the number of operating hours per year. For a hospital or other building running continuously (8,760 h/y), the paybacks (using 1989 prices) fall to 2.7 years for 0.62 kPa, and 1.7 years for 0.37 kPa.

For a new installation, there is little doubt that DDC zone control is more expensive than pneumatic. At the time of this writing, DDC boxes typically cost about $250-$275 more than pneumatic boxes, which cost about $300-$400 if fan-powered, and $80-$150 if not.[49] Installation is more expensive, requiring programming the zone management system and debugging the network.[50] Pneumatic terminal boxes, having many more moving parts, need at least yearly maintenance and adjustment; air compressors must be serviced

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[48]Typical costs were (EPRI, 1987):
7.5 - 50 hp: $900 - $425/hp
50 - 200 hp: $425 - $325/hp

[49](Fish, 1990, and Int-Hout, 1990)Accounting for the thermostat (about $50-$75), generally included with DDC boxes but not with pneumatic ones, the DDC box costs only $175-$225 more.

[50]Debugging the network—i.e., solving communications problems—may take one hour per box, depending on the skill of the installers.

regularly; the service requirements for a DDC system are considerably less. Adding typically standard DDC features like dual-minimum flow rate[51] to a pneumatic box raises the cost considerably. **Table 2.4** shows a comparison of estimated installed costs for pneumatic and DDC terminal boxes.[52] The quantities of each type of box used (e.g., fan powered, reheat, etc.) are the same as those used in the study building, for both cases. It is assumed that the rest of the control systems in the building are DDC, therefore the cost of the compressor can be attributed solely to the zone temperature control system, in the pneumatic case. The total system cost is $91,000 for the pneumatic system, and$112,000 for the DDC system, a difference of $21,000.

Assuming the DDC system enables the reduction of static pressure on the order of the example given in **Table 2.3**, an annual savings of $3,500 ($6,400 – $2,900) would result, in addition to $5,000 in avoided maintenance (calibration) costs,[53] for a total savings of $8,500 per year. A simple payback period of 21,000/8,500 = 2.5 years makes the DDC system a good investment, in new construction, primarily due to the lower cost of maintenance. Adding dual-minimum flow rate capability to the pneumatic boxes with reheat adds about $6,000 to the system.

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[51]This feature enables decreasing the minimum supply air flow rate when the zone is unoccupied, in order to reduce energy used for ventilation, heating and cooling.

[52]Costs shown are for a hot water reheat system; costs for a system with electric reheat were estimated as well, and are within 2% of those for the hot water system.

[53]Assuming negligible maintenance cost for the DDC boxes. According to manufacturers, this is a reasonable assumption.

Table 2.4. COMPARISON OF INSTALLED COSTS FOR PNEUMATIC AND DDC TERMINAL BOX SYSTEMS (Hot Water Reheat)

	Pneumatic				DDC			
	No fan		Fan Powered		No fan		Fan Powered	
Box	150	150	405	405	430	430	710	710
Thermostat	50	50	50	50				
HW Coil, valve, act.		85		110		115		140
Tot. Equipment	200	285	455	565	430	545	710	850
Fitter	100	150	100	150		50		50
Technician					20	20	20	20
Electrician			60	60	60	60	60	60
Balancer	40	40	40	40	20	20	20	20
Total each type	**340**	**475**	**655**	**815**	**530**	**695**	**810**	**1,000**
Quantity	3	19	36	67	3	19	36	67
Total Each Type	1,020	9,025	23,580	54,605	1,590	13,205	29,160	67,000
Compressor + labor				3,000				
Network & laptop								1,450
Total Installed Cost				**91,230**				**112,405**
Additional Features:								
Dual minimum (ea)		70		70		0		0
Total additional cost	0	1,330	0	4,690	0	0	0	0
Total				**97,250**				**112,405**
Maintenance:								
Calibration, 2x/yr	40	40	40	40				
Calibration, subtotal	120	760	1,440	2,680				
Calibration, per year				**5,000**				

Notes:
Source of price estimates: J. Fish, Titus Division of Philips, January 1990. Box prices include 0.18 multiplier, 20% markup. Fitter/electrician/balancer rate = $40/hr; technician rate = $20/hr. Quantities same as for Enerplex South. Fitter cost includes labor and materials for PVC pneumatic lines and hot water plumbing. For copper pneumatic lines, add $100 per box. Compressor includes dryer. Water piping & sheet metal work not included. Annual filter replacement on fan powered boxes not included.

2.6 Conclusions: Variable Speed Drives

One important result of the work described in this chapter is the quantification of the discrepancy in fan energy use as a consequence of applying power curves for systems that do not maintain constant duct static pressure to those that do. Indiscriminate use of such curves may be a natural consequence in a field where even the most authoritative literature (ASHRAE, 1988) and state-of-the-art building simulation model, DOE-2, have been unable to keep up with the rapid advance of technology. This illustration of the technology/information time lag makes a good case for improvements in this area.

The economics of installing VSDs point to their use in new construction rather than as a retrofit item, except in buildings that operate continuously, or under financial circumstances where a payback longer than 3 years is acceptable. The substantial energy savings possible with VSDs will not offset their high cost until economies in production can reduce this, which will happen as they become more popular.

Malfunctioning terminal boxes increase energy consumption (for VSDs or VIVs) and nullify retrofit energy saving. In light of this, it is clear that the whole air distribution system must be checked periodically for such failures, and VSD retrofit jobs should not be given final approval until problems such as these have been corrected.

For the VSDs, the magnitude of the additional annual savings due to lowering duct static pressure (164 MWh - 101 MWh = 63 MWh) is two thirds of the savings resulting from the variable speed drives alone (101 MWh), while lowering duct static pressure in a VIV system has little benefit, indicating a synergistic relationship between the two measures. Manually resetting duct static pressure, a low-cost measure, is not feasible if frequent adjustments are required. While the combined savings are probably not high enough to justify the cost of a control system retrofit that would enable automatic duct static pressure minimization, they do bolster the feasibility of VSDs in conjunction with such a control scheme in new construction where digital control is used.

We have only briefly explored the question of how varying duct static pressure set point affects energy use in a VIV system; large uncertainties in the data make it difficult to discern a subtle influence. If these results can be generalized, however, they have important implications for ventilation system control—duct static pressure reset in a VIV system is not worth any added expense unless it is expected that VSDs will be installed at a point in the future.

Precise duct static pressure control can not be achieved without the addition of an electronic control system, which permits communication with each of the terminal boxes to ensure that none of them have insufficient inlet pressure, and with terminal boxes capable of providing minimum ventilation air at low static pressures while in heating operation. Such a control system is not at all common in buildings today, and is in fact just becoming possible with the introduction of DDC terminal boxes (although conceivably a system could simply use switches to determine whether any boxes were starved for air). The economic comparison of DDC vs. pneumatic terminal boxes presented here shows DDC boxes to be a favorable investment for new construction if viewed from a life-cycle cost perspective, as the lower operating and maintenance cost relative to a pneumatic system offsets the higher initial capital cost.

In Chapter 3 we will focus on technical details of VAV zone temperature control, contrasting pneumatic and digital systems, and strategies for improving control using new technologies. We will continue the discussion of static pressure minimization in Chapter 4, with simulations of the supply air system and energy efficient control techniques.

Chapter 3

Digitally Controlled Terminal Boxes and Improved Zone Temperature Control

Chapter 2 brought out several limitations of pneumatic terminal box control, primarily due to the inability of boxes to communicate with the main fan control system:

- Manual static pressure adjustment could not be implemented without risking overheating zones during the cooling season or under-ventilating them during the heating season.
- It is impossible to implement advanced fan control strategies employing terminal box feedback, such as static pressure minimization.
- It is difficult to identify boxes that are not balanced in accordance with the loads on a zone. Flow set points may not be high enough to meet cooling load, resulting in overheating. It is also impossible to change flow set points in a non-intrusive manner.
- It is difficult to identify malfunctioning terminal boxes.
- It is difficult to identify undersized boxes (i.e., boxes already adjusted to their flow capacity), which may result in overheating during the cooling season.

These limitations can result in sub-optimal energy efficiency and comfort. In this chapter, and the one following, we will explore some of the improvements that are possible with digital control of VAV systems, in particular, those that combine the advantages of variable speed drives with digital zone temperature control.

We approach the question of improving the control of the supply fan and terminal box from two directions: 1) an experimental study of DDC terminal box and office space response to changes in static pressure and internal heat gain, and 2) computer simulations of this system. In order for this approach to be meaningful to the reader, we must first develop a physical understanding of the principles underlying zone temperature control and the equipment involved. Although this may seem tedious, the reader's patience will be rewarded by a better appreciation of the subtleties of control that will arise later in this chapter and in Chapter 4.

In Section 3.2, we describe experiments conducted on a test installation of a DDC terminal box in a commercial office space, and present the results of trials performed to determine the effect of changing duct static pressure and internal gains on terminal box and zone temperature control. We explain, in Section 3.3, the development of the terminal box and zone model, and describe each of the component models and the procedures used to identify parameter values based on measurements of the experimental DDC system. We then present a validation of the complete model using experimental data. Finally, in section 3.4, we discuss some of the other improvements in zone temperature control made possible by DDC terminal boxes.

We begin with a discussion of the fundamental thermal dynamics involved in zone temperature control, followed by a detailed look at terminal box dynamics, and a comparison of pneumatic and DDC terminal box controllers.

3.1 ZONE TEMPERATURE CONTROL

ZONE DYNAMICS

In order to understand the principles behind the design of a terminal box controller, it is useful to look at the terminal box and zone as a system, beginning with the equations for conservation of energy and continuity. **Figure 3.1** shows a schematic section of the terminal and zone. An energy balance on the air in the zone can be approximated as:

$$\rho V_r c \frac{dT_r}{dt} = \rho \dot{V}_o c T_o - \rho \dot{V}_o c T_r + \dot{Q}_l$$

or

$$\frac{dT_r}{dt} + \frac{\dot{V}_o}{V_r} T_r = \frac{\dot{V}_o}{V_r} T_o + \frac{\dot{Q}_l}{\rho c V_r} \qquad (3.1.1)$$

where

ρ	=	density of air, taken as 1.1 kg/m^3
V_r	=	volume of air in the zone
c	=	specific heat of air, taken as 1.0 kJ/kg-C
T_r	=	temperature of air in the zone
$\dot{V}_o$	=	flow rate of air from the outlet of the terminal box into the zone
T_o	=	temperature of the air stream at the outlet of the terminal box

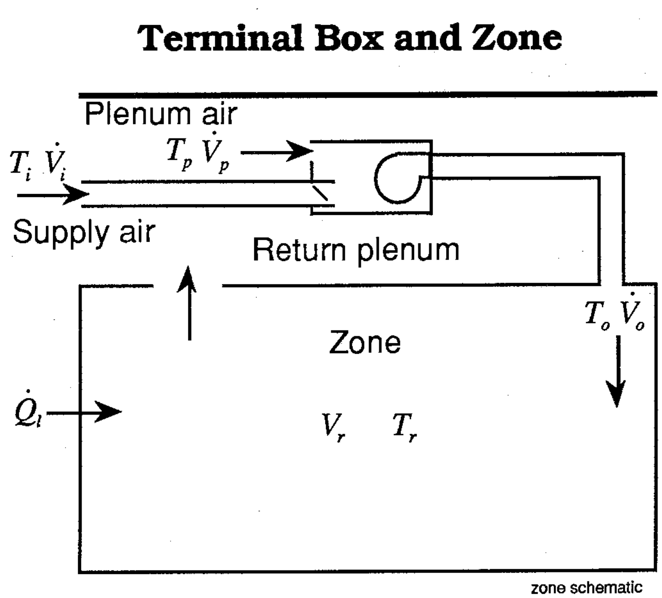


Figure 3.1. *Schematic section of the terminal box, ceiling plenum, and zone.*

$\dot{Q}_l$ = heat gains (losses) to the zone, including internal gains from lights, people and equipment, solar gains through glazing, conductive transfer through the zone envelope from outside and adjacent spaces, and convective transfer from the material surfaces in contact with the room air.

It is assumed that the temperature in the zone is uniform, i.e., the air is well mixed,[1] that the air is dry, and density and specific heat are constant.[2] The gain term $\dot{Q}_l$ includes the effect of heat stored in the walls, floor, furniture, and other surfaces in contact with the air. Temperature of this thermal mass would be included as a separate equation in a complete model.[3]

Terminal Box Dynamics

Neglecting reheat and energy added by the fan, an energy balance on the terminal box yields

$$\dot{V}_p T_p + \dot{V}_i T_i = \dot{V}_o T_o \tag{3.1.2}$$

where

$\dot{V}_p$ = flow rate of (secondary) air through plenum inlet of box

T_p = ceiling plenum air temperature

$\dot{V}_i$ = primary air flow rate

T_i = primary air temperature

Again, the assumption of constant density and specific heat is made. The equation for continuity through the box is

---

[1]While good air movement has been observed in offices in the study building, it may take a few minutes for the air leaving diffusers to become well mixed in the zone. Hence the validity of this assumption depends on the time scale of interest.

[2]Constant density and specific heat are reasonable assumptions for the temperature range of interest.

[3]Heat exchange between the thermal mass and room air has been lumped together with the other gains and losses not because it is insignificant, but its inclusion here would complicate the discussion unnecessarily and would not add significantly to the basic understanding of zone temperature control.

$$\dot{V}_o = \dot{V}_p + \dot{V}_i \tag{3.1.3}$$

Substituting (3.1.3) into (3.1.2) and eliminating $\dot{V}_p$, we have

$$T_o = T_p - V_i\left(\frac{T_p - T_i}{\dot{V}_o}\right) \tag{3.1.4}$$

Constant volume boxes are designed to deliver a constant flow $\dot{V}_o$. Assuming that T_p and T_i are constant as well, the outlet temperature is thus proportional to the primary inlet flow rate. In cooling mode, zone temperature is controlled by modulating the damper which changes $\dot{V}_i$ hence changing T_o.[4]

A block diagram of the terminal box/zone system is shown in **Figure 3.2.** The system can be broken down into an inner loop, which controls primary flow rate, and an outer loop, which controls zone temperature. Existing terminal box controllers, whether pneumatic or DDC, are based on this scheme, using proportional-only control for the outer loop. In the inner loop, the position of the damper is modulated to regulate the measured primary air flow rate, $\dot{V}_i$, about the desired flow rate, $\dot{V}_{des}$, which in turn is proportional (within a throttling range) to an error signal that is the difference between the zone temperature measured at the thermostat and the zone temperature set point. $\dot{V}_{des}$ varies between a minimum, $\dot{V}_{min}$ based on a ventilation requirement, and a maximum, $\dot{V}_{max}$, based on a design cooling load.

[4]Boxes without reheat vary outlet temperature only in cooling mode. See Section 1.2.

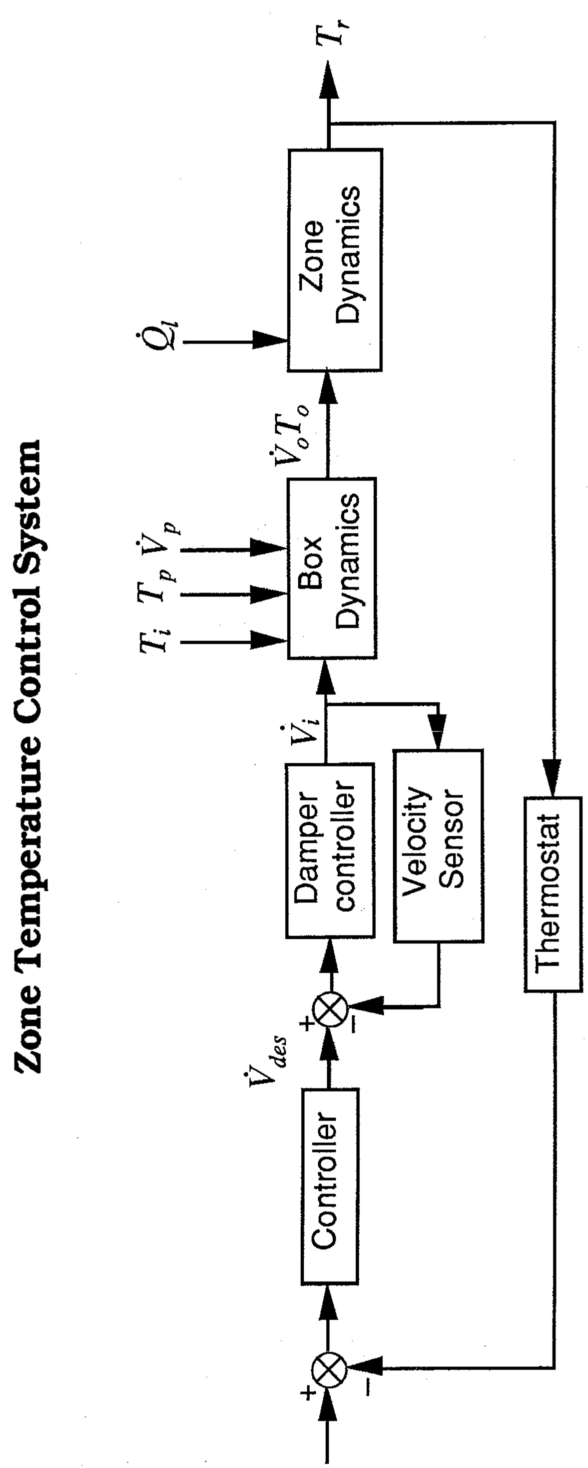


Figure 3.2. *Block diagram of terminal box and zone temperature dynamics. The inner loop regulates the flow rate to a setpoint determined by the outer loop, which regulates the air temperature of the zone.*

box contr blkdiag

Pneumatic and DDC boxes differ fundamentally, however, in how they regulate flow rate. Pneumatic boxes use proportional control for the inner loop, whereas DDC boxes use a form of proportional plus integral (PI) control here.[5] A pneumatic box will therefore exhibit a steady state error in the flow rate, its magnitude varying with the primary inlet duct pressure; DDC boxes are more truly "pressure-independent." This is discussed in more detail below.

In terminal boxes with reheat, as zone temperature drops below set point, a reheat actuator is operated in a proportional fashion such that the amount of heat added at the box outlet is proportional to the zone temperature error signal. Hot water reheat systems do this by modulating a valve; electric systems energize resistance elements in stages. The control sequence for a typical constant volume terminal box with hot water reheat is shown in **Figure 3.3.**

THE PNEUMATIC CONTROLLER

The transient response (Clark, 1985) of a PI controller can be expressed as

$$\frac{dC_{out}}{dt} = \frac{C_{out}^{ss} - C_{out}}{\tau} \tag{3.1.5}$$

$$C_{out}^{ss} = K_P E + I \tag{3.1.6}$$

$$E = X_{set} - X_{meas} \tag{3.1.7}$$

$$\frac{dI}{dt} = K_I E \tag{3.1.8}$$

where

C_{out} = controller output

[5] Integral control is cumbersome (and expensive) to implement pneumatically. The DDC terminal box used in these experiments controlled the flow rate to set point using a heuristic "step and wait" method, giving the effective result of an incremental form of discrete time PI control.

C_{out}^{ss} = steady-state value of controller output

τ = time constant of controller response

K_P = proportional gain

I = integral portion of the output signal

E = controller error

X_{set} = set point for controlled variable X

X_{meas} = measured value of controlled variable X (feedback signal)

K_I = integral gain, s^{-1}

A more common form of Equations 3.1.6 through 3.1.8 is (Ogata, 1970)

$$C_{out}^{ss} = K_P\left(E + \frac{1}{t_I}\int_0^t E dt\right) \tag{3.1.6b}$$

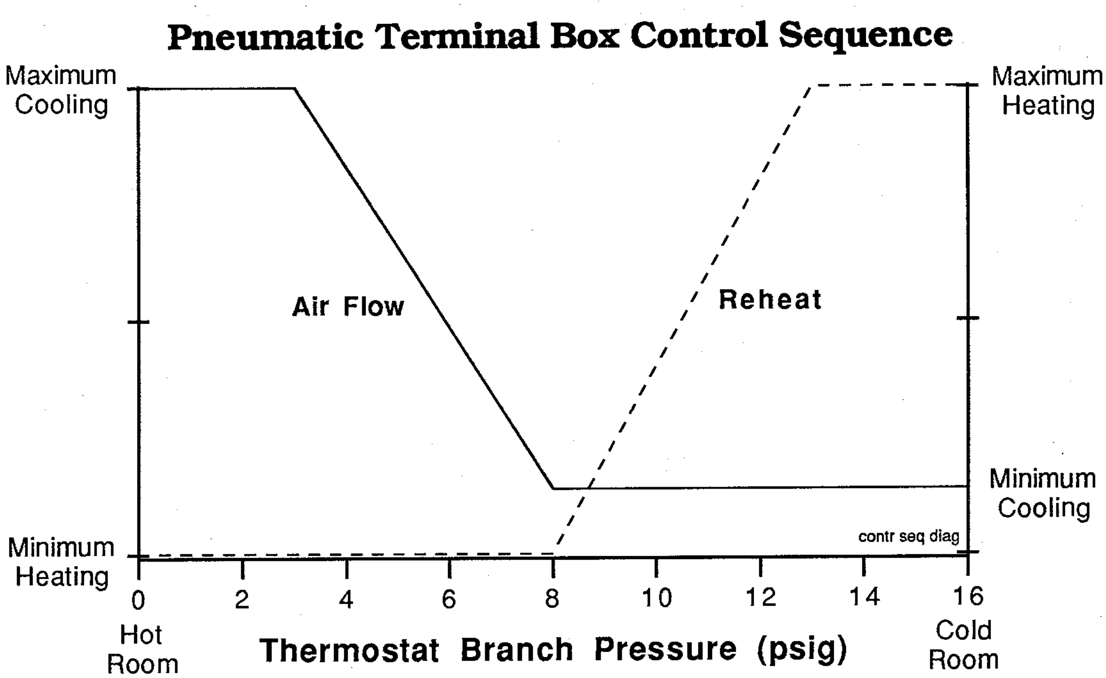


Figure 3.3. Control sequence for pneumatic terminal box. The thermostat branch pressure is proportional to the room temperature minus set point. In heating mode, as the room cools down, the damper closes to a minimum, and the reheat valve opens to a maximum. In cooling mode, the damper opens as the room heats up.

where

t_I = integral time, s (= K_P / K_I)

With K_I set to zero, this reduces to a proportional controller with offset, where the integral term I becomes the zero offset term. The steady-state equation for the output of a such a controller is thus

$$C_{out}^{ss} = K_P E + C_{zo} \tag{3.1.9}$$

where

K_P = proportional gain $= \dfrac{R_{out}}{R_t}$

R_{out} = controller output range (maximum – minimum output)

R_t = throttling range

C_{zo} = zero offset

The zero offset, which is the output of the controller at zero steady-state error, is typically set equal to half the maximum output, so that the throttling range is centered on the value of the controlled variable requiring 50% of the output of the controller (Nesler and Stoecker, 1984).[6] The output of the controller typically has some finite range.

The pneumatic zone temperature controls in the study building act in this manner. Based on field measurements (**Figure 3.4**), thermostats have an output range of 0 to 96 kPa (0 to 14 psig), a throttling range of about 2.2 °C (4 °F), a zero offset of 41 kPa (6 psig), and are reverse acting. Represented in the form of Equation 3.1.9, this gives K_P = –96 kPa/2.2 °C = –43.4 kPa/°C (–3.5 psig/°F), and C_{ZO} = 41 kPa (6 psig) (**Figure 3.3**):

$$C_{stat}^{ss} = -43.4(T_{set} - T_r) + 41 \tag{3.1.10}$$

[6]The minimum controller output is sometimes chosen as the zero offset.

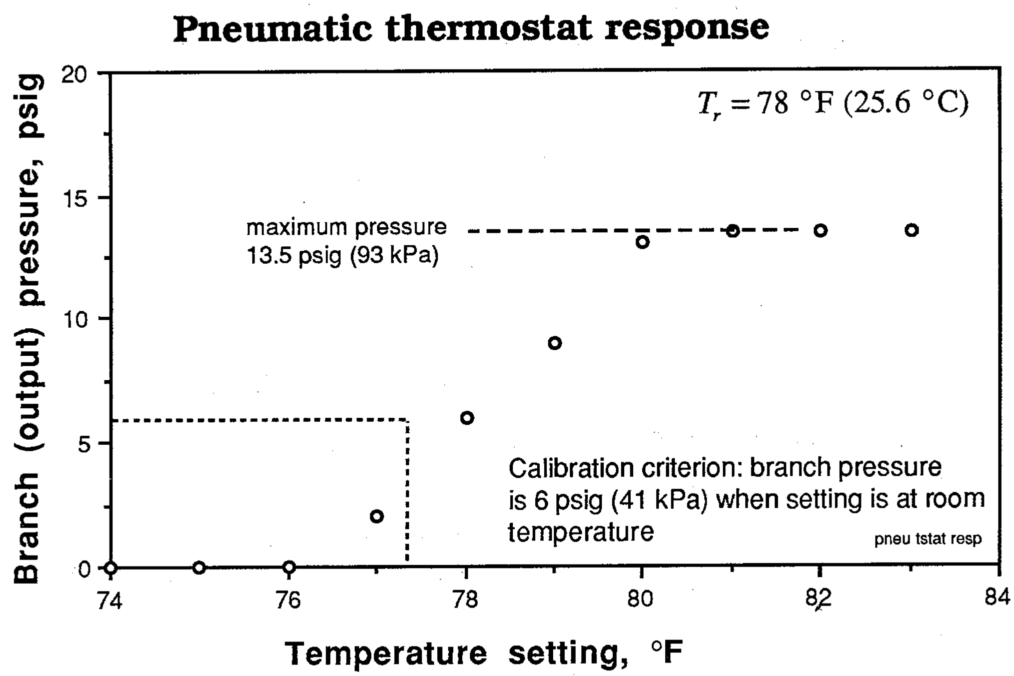


Figure 3.4. Measured response of pneumatic room thermostat to changes in set point, indicating proportional band.

where

T_{set} = temperature set point, °C

T_r = room temperature at location of thermostat, °C

and C^{ss}_{stat} is in kPa. The proportional action of the thermostat is thus roughly centered about the 50% output point. The output signal C^{ss}_{out} is proportional to a desired primary air flow rate $\dot{V}_{des}$, and serves as the set point input to the flow rate controller.

The pneumatic flow controller (the inner loop of **Figure 3.1**) in the study building has a reset range of 21 kPa to 55 kPa (3 psig to 8 psig),[7] and is normally open. This range, then, corresponds to an effective throttling range of about 0.83 °C (1.5 °F), from 0.33 °C (0.6 °F) below to 0.5 °C (0.9 °F) above set point. The controller regulates the flow rate using proportional control of the damper. Ideally, the actual flow rate, when equal to

[7]In boxes with reheat, the 55 kPa to 90 kPa (8 psig to 13 psig) range is reserved for the heating actuator, i.e. the normally closed valve controlling hot water flow through the reheat coil.

the desired flow rate, should be proportional to the difference between temperature set point and measured room temperature.

In heating mode (when zone temperature is below set point and the primary air is at its minimum), the hot water valve on a reheat box modulates similar to the air flow rate control, as in **Figure 3.3.** The reset range of the actuator is 55 kPa to 90 kPa (8 psig to 13 psig), which corresponds (using Equation 3.1.10) to an effective throttling range of 0.78 °C (1.4 °F), from 0.33 °C (0.6 °F) below set point to 1.1 °C (2.0 °F) below set point. In the study building, operating this valve has no effect when the building is in "summer" mode, as there is no hot water in the loop.

The flow rate feedback signal in the pneumatic controller is produced by a dynamic pressure probe, and is proportional to the square of the velocity (and hence to the square of the volumetric flow rate $\dot{V}$).[8] If the damper position were determined by the output of a proportional controller using as inputs the thermostat output (as set point) and this feedback signal,[9] it would take the form

$$C_d = K_4\left[K_1\left(T_{set} - T_r\right) + K_2 - K_3\dot{V}^2_{meas}\right] + K_5$$

which reduces to

$$C_d + K_6\dot{V}^2_{meas} + K_7 = K_8\left(T_{set} - T_r\right) \tag{3.1.11}$$

where

C_d = controller output to the damper actuator, 0 = closed

$\dot{V}_{meas}$ = measured primary air flow rate, m^3/s or cfm

and the K are constants. Thus, in order for flow rate to be proportional to the temperature difference (within a proportional band), the characteristic of the damper would have to match that of the flow sensor/transducer, i.e.,

[8]As shown in manufacturer's literature for the "EconoFlo" terminal box (Barber-Colman).
[9]This assumes a linear actuator. The actuator linkage in an actual terminal box results in a slightly non-linear action.

$$C_d = -K_6 \dot{V}_{meas}^2 + K_9 \dot{V}_{meas} + K_{10} \tag{3.1.12}$$

$$\dot{V}_{meas} = K_{11} + K_{12}(T_{set} - T_r) \tag{3.1.13}$$

This damper characteristic is of the inverse parabolic or "quick opening" type (Letherman, 1981). Although in this study, no measurements of flow rate versus damper position for pneumatic boxes (which have 90° dampers) were made, such measurements were taken for a DDC box (which has a 45° damper) over a range of static pressures (**Figure 3.5**). Qualitative visual analysis indicates an installed characteristic of this type for the 45° damper; it is not known whether the installed characteristic of the 90° damper is such that flow rate varies linearly with temperature difference.

As is evident in the **Figure 3.5**, static pressure has an enormous effect on flow rate. Thus for a pneumatic box, as Pinnella (1985) points out, the flow response will not be linear with temperature difference (as in **Figure 3.3**) unless the inlet pressure is equal to the design pressure, i.e., that pressure for which the flow vs. temperature response is linear. Static pressures below this effectively decrease the gain in the temperature loop, perhaps resulting in slow response to step changes in cooling load; higher static pressures would effectively increase the gain , which might result in greater oscillation in zone temperature, or (in the heating season) excessive reheat coil energy use.

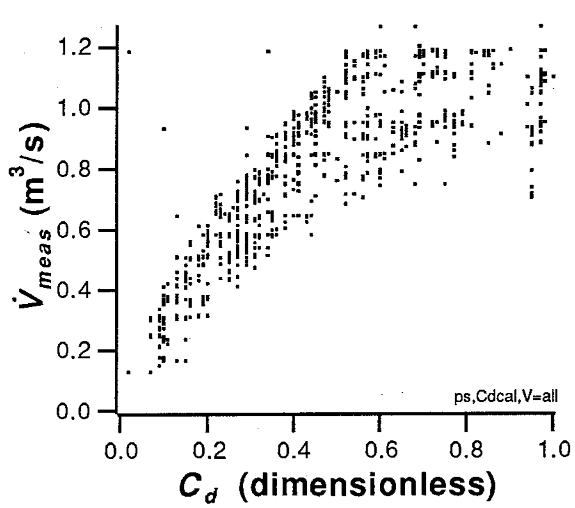


Figure 3.5. Installed damper characteristic for 45° damper in DDC terminal box. Measurements of flow rate at a given damper position increase with static pressure.

THE DDC CONTROLLER

The DDC terminal box used in this study uses an RTD sensor in the zone thermostat to sense temperature;

the position of a potentiometer indicates local temperature set point. The controller loops through a sequence of operations that includes

- sampling the sensor, potentiometer, and flow sensor
- analog to digital conversion
- computing a desired flow rate based on the difference between zone temperature and set point, and a damper control signal based on the difference between measured and desired flow rate
- outputting a signal to the damper actuator
- communicating with the supervisory control system via the network

Each iteration of the loop takes about 10 s to complete.

This computation of desired flow rate has the form of Equation 3.1.9 (steady-state proportional control with offset), except for two differences: the throttling range begins at set point, rather than straddling it, and there is an additional offset term which results in a dead band. The zero offset is then the minimum desired flow rate:

$(T_{set} + C_{oc} \le T_r \le T_{set} + C_{oc} + R_t)$

$$\dot{V}_{des}^{ss} = K_P(T_{set} + C_{oc} - T_r) + \dot{V}_{min} \tag{3.1.14a}$$

$(T_r \ge T_{set} + R_t + C_{oc})$

$$\dot{V}_{des}^{ss} = \dot{V}_{max} \tag{3.1.14b}$$

$(T_r \le T_{set} + C_{oc})$

$$\dot{V}_{des}^{ss} = \dot{V}_{min} \tag{3.1.14c}$$

$$K_P = \frac{\dot{V}_{max} - \dot{V}_{min}}{R_{tc}} \tag{3.1.15}$$

where

K_P = proportional gain, < 0

C_{oc} = offset, cooling (°C or °F), > 0

$\dot{V}_{min}$ = minimum flow set point (m^3/s or cfm)

$\dot{V}_{max}$ = maximum flow set point (m^3/s or cfm)

R_{tc} = throttling range, cooling (°C or °F), > 0

Since the output of this controller serves as the input to a mechanical actuator, it is desirable to lessen the frequency of stops, starts, and reversals, as well as "hunting" or oscillation, which cause wear and tear; the dead band serves this purpose.

To regulate the flow rate, the DDC terminal box used in this study controls the inlet damper using a discrete-time heuristic or "step and wait" process that has the effect of controlling flow to set point (in contrast to the pneumatic controller), as does PI control (Farkash, 1989). The damper is actuated by synchronous motor, which is pulsed with DC signals output by the controller to give the desired rate of travel. The damper motor is pulsed to run at full speed except when the error in flow rate is less than a threshold value (the "slow band", in which case the motor runs at half-speed so as to lessen overshoot and enable longer sampling intervals.

The reheat valve in the DDC system used here can be one of two types: a "floating" or pulsed actuator similar to the damper actuator, or a modulating actuator which takes a standard 4 mA to 20 mA control signal as input.

3.2 Exploring the Feasibility of Improved Control: Field Tests

At the time this study was initiated, DDC terminal boxes were just becoming commercially available, and there existed only a few buildings in the U.S. in which they were installed. To this day, there are few, if any, published reports on results of *in situ* studies of their performance. Experimental measurement of terminal box performance was therefore a key element of this study, with the following objectives:

- Observe the behavior of a DDC terminal box in conjunction with a VAV system and VSDs,
- investigate the feasibility of proposed control strategies, and
- make measurements of operational parameters for the purpose of simulation

In order to do this, an existing pneumatically controlled constant volume terminal box in an unoccupied but finished office space in the study building was replaced with a DDC box.

The box and zone were instrumented and tied into the existing data acquisition system. Manufacturers' data for the equipment are provided in Appendix B, and a schematic diagram of the box is shown in **Figure 3.6**. Appendix A gives all instrumentation details, which are summarized below in **Tables 3.1** and **3.2**, listing the data that were measured.

The zone consists of four perimeter offices on the east side of the ground floor of the building, a total of 70 m^2 (755 ft^2) (**Figures 3.7** and **3.8**). This zone was selected for experimental purposes because unlike most others in the building, it is served by only one terminal box, and this terminal box serves only this zone. An architectural section in **Figure 3.8** shows construction details. **Figure 3.9** shows the interior of one of the offices, and the terminal box installed.

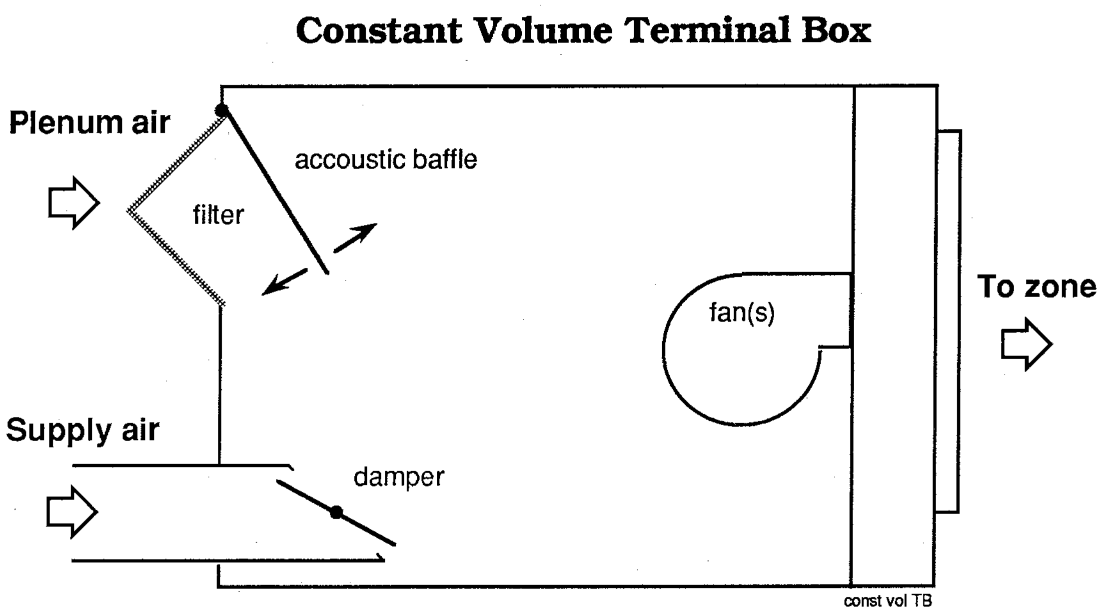


Figure 3.6. Schematic of the constant volume terminal box used in the experiments. This type of box has one or two fans, depending on size. The accoustic baffle is a piece of thin sheet metal which swings freely in the plenum air inlet, and is closed when the supply air flow is at a maximum, i.e., equal to the rate the fan is adjusted to produce. The filter was not in place during the experiments.

Table 3.1. TERMINAL BOX INSTRUMENTATION (Power Line Carrier System)

Channel	Description	Location	Sensor Type
Tr1	room temperature	office 1	RTD
Tr2	room temperature	office 2	RTD
Tr3	room temperature	office 3	RTD
Tr4	room temperature	office 4	RTD
Tr5	room temperature	interior zone	RTD
L1	light status	office 1	CdS photocell
L2	light status	office 2	CdS photocell
L3	light status	office 3	CdS photocell
L4	light status	office 4	CdS photocell
D1	door status	office 1	magnetic switch
D2	door status	office 2	magnetic switch
D3	door status	office 3	magnetic switch
D4	door status	office 4	magnetic switch
Ti	inlet air temperature	terminal box	RTD (thin film)
To	outlet air temperature	terminal box	RTD (thin film)
Tp	plenum temperature	terminal box	RTD (thin film)
Cd	damper position	terminal box	precision pot.

Table 3.2. ADDITIONAL INSTRUMENTATION (Terminal Box Network and Main DAS)

Channel	Description	Location	Sensor Type
Tr2t	temp. at thermostat	office 2	RTD
Vdes	desired zone flow rate	box controller	—
Vmeas	measured zoneflow rate	primary inlet	dyn. pres. sensor
Vmaxdy	max daytime flow rate	box controller	—
Vmindy	min daytime flow rate	box controller	—
Tset	temperature set point	office 2	potentiometer
Fan	term. box fan status	box controller	—
Mode	terminal box mode	box controller	—
ps2	AC2 duct static pres.	control panel	0-25 psig transducer
Vs2	AC2 supply air flow	basement duct	TAMS[a]
Ts2	AC2 suppl. air temp	betw. coil , fan	Bourdon tube
AC2kWh	AC2 fan electricity	basement meter	pulse head on meter
Qsolt	total horizontal solar rad	roof	pyranometer
Tamb	outside temperature	outside	thermistor

[a]thermistor air measuring system (multipoint)

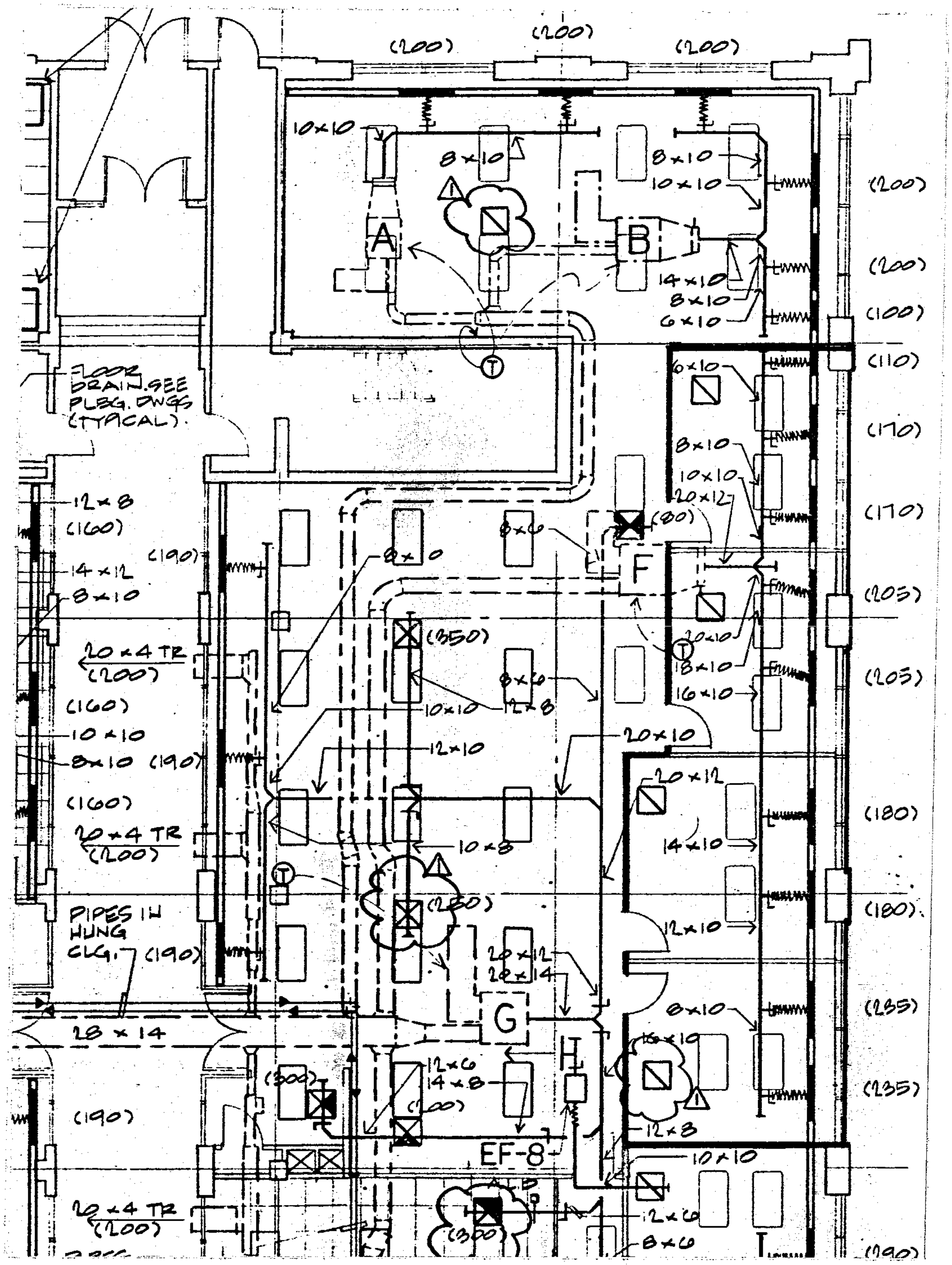


Figure 3.7. Mechanical plan of the four instrumented offices and surrounding area (outlined at the right). The windows face East. The terminal box labeled "F" is the one replaced. Detail shown in Figure 3.8. Source: Haines Lundberg Waehler, New York, N.Y., (for Dow Jones) drawing M-1, revised 8/20/84.

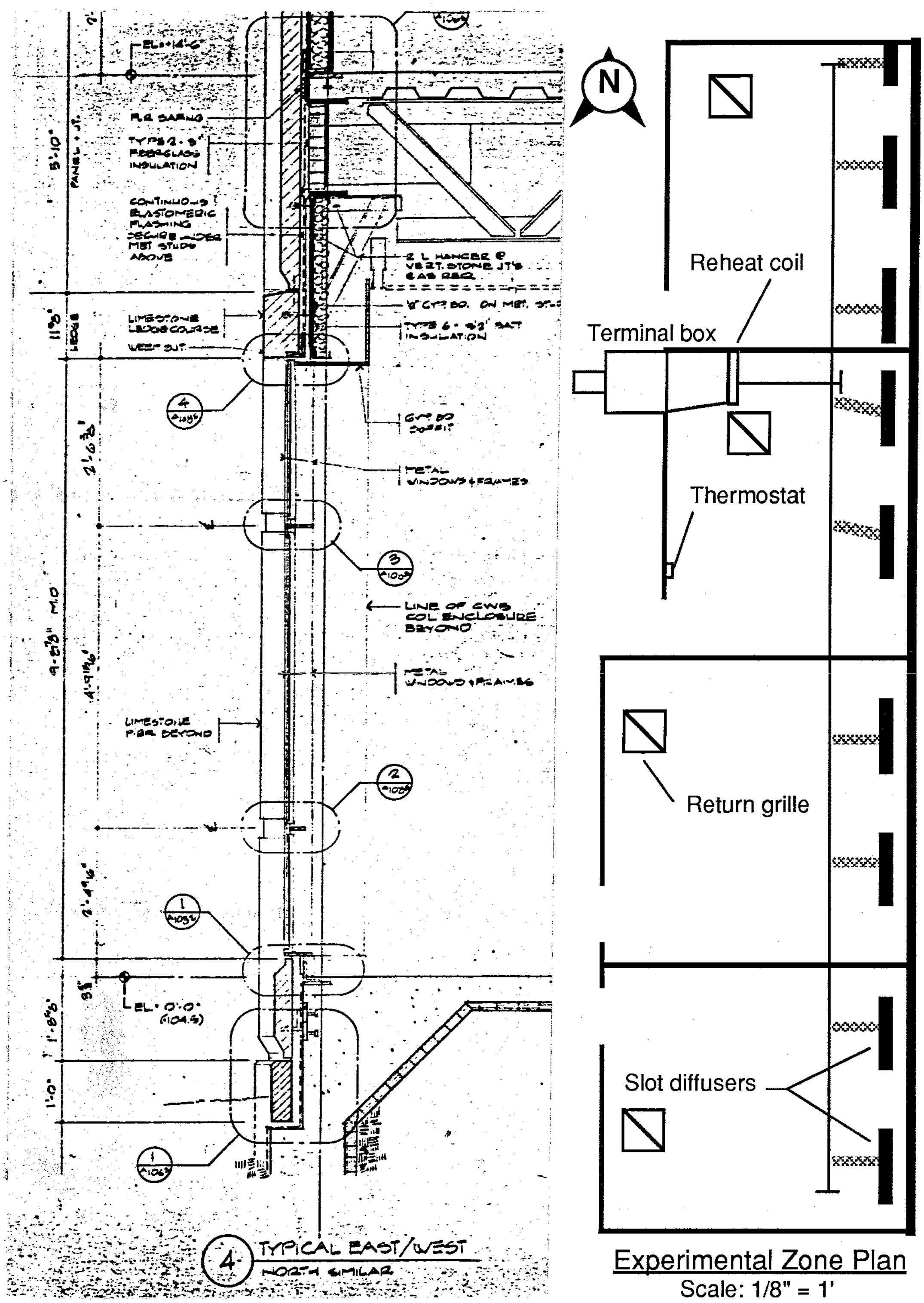
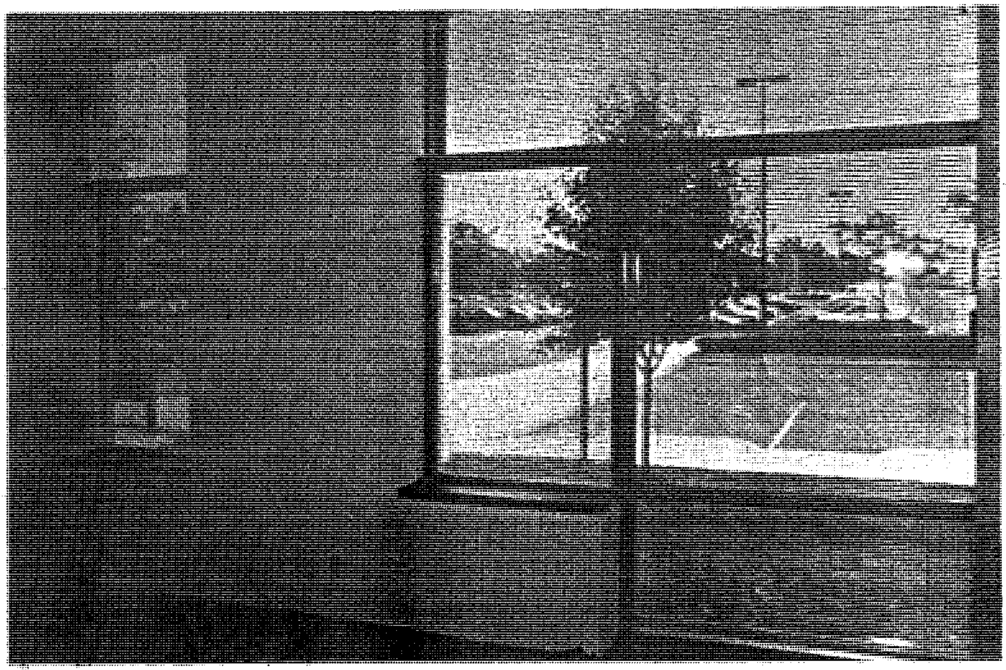


Figure 3.8. Wall and floor section showing construction details; plan of experimental office space (detail of Figure 3.7).



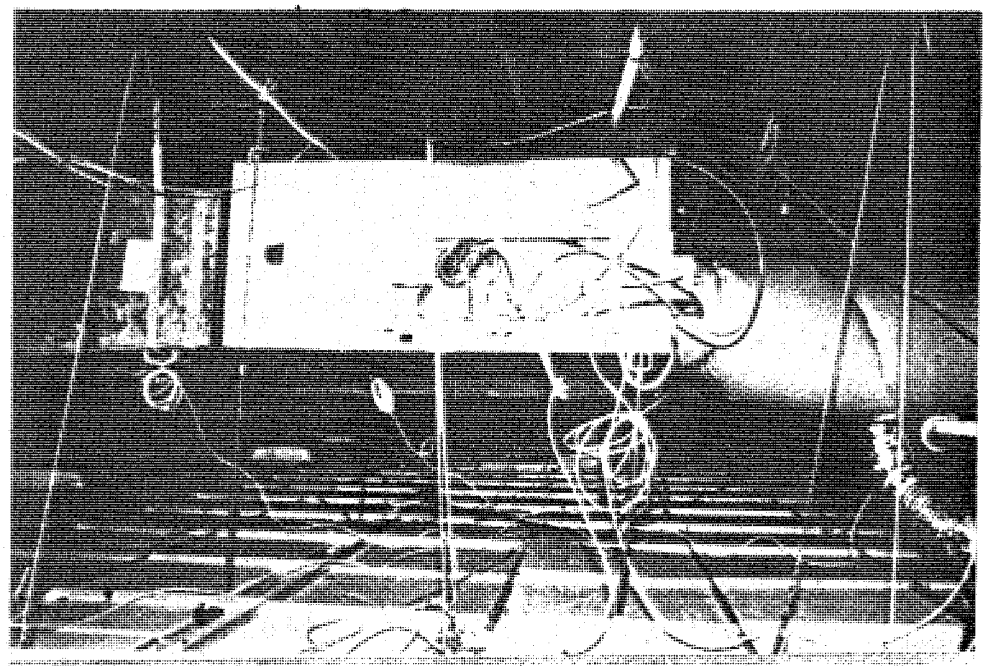


Figure 3.9. *Interior of office, facing East (above), and instrumented DDC terminal box, installed in ceiling plenum above unoccupied office space.*

EXPERIMENTAL INSTALLATION

All of the control functions performed by conventional non-DDC terminal boxes are handled by a microcontroller board mounted on the box; functions such as scheduling and mode changes are done remotely by the zone management system. The zone management system tells the controller on the box which of 12 modes[10] it should be in; the controller's actions are determined by the set of operational parameters defined for the current mode and current state (heating or cooling).[11] These parameters include minimum and maximum primary air flow rates, throttling range and offset, and remote temperature set points. The operational parameters of the experimental terminal box are stored in EEPROM on its controller board.

The parameters were configured on site through software either by connecting a portable computer to a serial interface located at the wall thermostat, or remotely through the front-end computer and zone management system. Minimum flow was set at 0.17 m^3/s (360 cfm), based on the standard requirement of 0.0095 m^3/s (20 cfm) of fresh air per occupant[12] for four occupants, under "worst case" conditions for fresh to supply air ratio. Several values of maximum flow rate were used during the course of the experiments, ranging from 0.80 m^3/s to 1.27 m^3/s (1700 cfm to 2700 cfm), almost the maximum the fan could deliver. Factory-set values for cooling mode throttling range (1.7 °C or 3 °F) and offset (none) were used. The fan speed was adjusted so that the flow at the box outlet was equal to the desired maximum. Since a diffuser flow measurement hood was not available, the fan speed was adjusted until the pressure difference across the secondary air inlet was

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[10]The number and specific nature of modes varies among manufacturers; for the equipment used in this study and described in Appendix B, there are six dynamic modes—warmup, three daytime modes, night unoccupied, and night occupied—and six static modes: smoke purge, fan purge, vacant, and three unconfigured (i.e., available) static modes. In the static modes, sensor inputs have no effect on control action. Some manufacturers use the terms "mode" and "state" in the sense opposite to that used here.
[11]DDC terminal boxes may be configured with a dead band between heating and cooling states; some controllers treat this as a third state.
[12]ASHRAE standard 62-1989; amount used is standard for office spaces.

approximately zero with the primary air flow at its desired maximum. This point was determined by observing the motion of the acoustic baffle hanging in the plenum inlet.

All data were recorded at one-minute intervals, except for outdoor ambient temperature and solar radiation, which were recorded at ten-minute intervals.

EXPERIMENTAL STRATEGY

In order to accomplish the objectives outlined above, it was necessary to perform a number of trials, each several hours long, during which the damper position, flow rate, and room temperature would be measured, with static pressure varied manually and step changes in internal gain introduced periodically to simulate office equipment. During some of the trials, static pressure would be adjusted manually to keep the damper open as wide as possible (to simulate static pressure minimization), while still providing the required flow rate, as determined by the terminal box controller. Loads would be introduced over a range of static pressures and damper positions. In addition, measurements would be taken of key variables required to calibrate the component models of the fan, damper, ducts, controllers, actuator, sensors, and room.[13]

FIELD TESTS

Several field tests were performed with the DDC equipment described above, resulting in sets of one-minute data for periods of several hours; the results of two experiments are presented here. Results of the first experiment, in which the response of the system to changes in duct static pressure p_s and internal gain were observed over a five hour period, are shown in **Figures 3.10** and **3.11** ("Experiment 1"). Static pressure was varied by manually using the pneumatic set point adjustment for the supply fan AC-2. Although some attempt was made to keep the terminal box damper open as wide as possible, this

[13]Calibration of the component models is discussed under "The Component Models," following.

proved to be too difficult, as only measurements of primary air flow $\dot{V}_i$ and desired flow rate $\dot{V}_{des}$ (not relative damper position C_d)were available at the location of the controller. In Experiment 2 (**Figures 3.12** and **3.13**), static pressure is left constant at two set points, 0.63 kPa (2.5 in. H_2O) and 0.26 kPa (1.1 in. H_2O).

Space heaters were used to introduce step changes in internal gain in the modeled room, where the thermostat was located; lights were the only source of internal gain in the three remaining rooms. In addition to 200 W due to lights, two levels of added gain were used—1650 W and 3150 W, for total peak levels (including lights) of 114 W/m^2 and 207 W/m^2; the average internal gain (including lights) for Experiment 1 was 1830 W, or 113 W/m^2; the average for Experiment 2 was 1820 W, or 112 W/m^2. The levels chosen are intentionally high compared to average office loads, so as to bring about changes in room temperature faster than would normally be seen, hence pushing the terminal box to its limit, and also to observe the minimum duct static pressure feasible under high loads. For comparison, a highly-loaded office containing a personal computer, laser printer, photocopier, and coffee brewer might have an average load of 1870 W, with a peak of 4520 W.[14] Typical peak equipment loads for mixed-use commercial buildings are around 14 W/m^2 (1.31 W/ft^2).

The maximum flow rate for both experiments shown here is 1.18 m^3/s (2500 cfm) or 0.017 m^3/s (3.3 cfm/ft^3) normalized by floor area. There was no direct solar gain through the glazing over the (afternoon) period of Experiment 1, as the windows faced East; morning sun was a factor in Experiment 2. The temperature set point was 70 F throughout both experiments.

[14]IBM PS-2/60, peak and average: 170 W (Norford et al., 1988); Apple LaserWriter Plus, peak: 230 W, average: 130 W (Norford et al., 1988); Konica 7090s copier (16 A, 220 V, power factor 0.7, use factor 0.3), peak: 2460 W, average: 740 W; Brewmatic M-80 (1660 W, use factor 0.5), peak: 1660 W, average: 830 W.

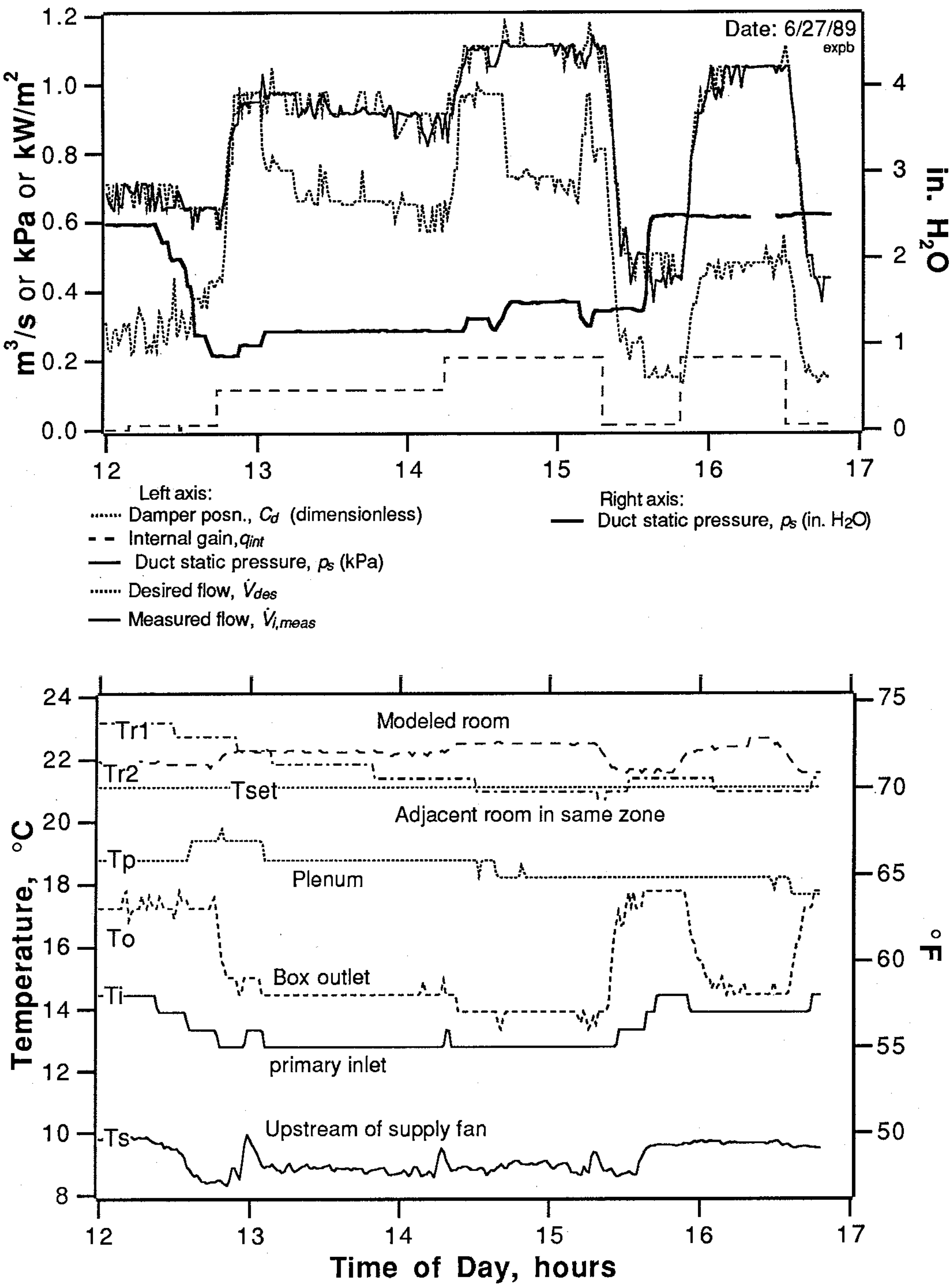

Figure 3.10. Measured response of terminal box and zone to manual changes in static pressure p_s and internal gain (Experiment 1). Space heaters were used to introduce step changes in internal gain. There was no direct solar gain during the period of the experiment. The first graph illustrates the time scale of the exponential response of temperature (and hence flow) to step changes in gain, about 10 minutes. The response of damper position to a change in static pressure can be clearly distinguished in the vicinity of 15 hours.

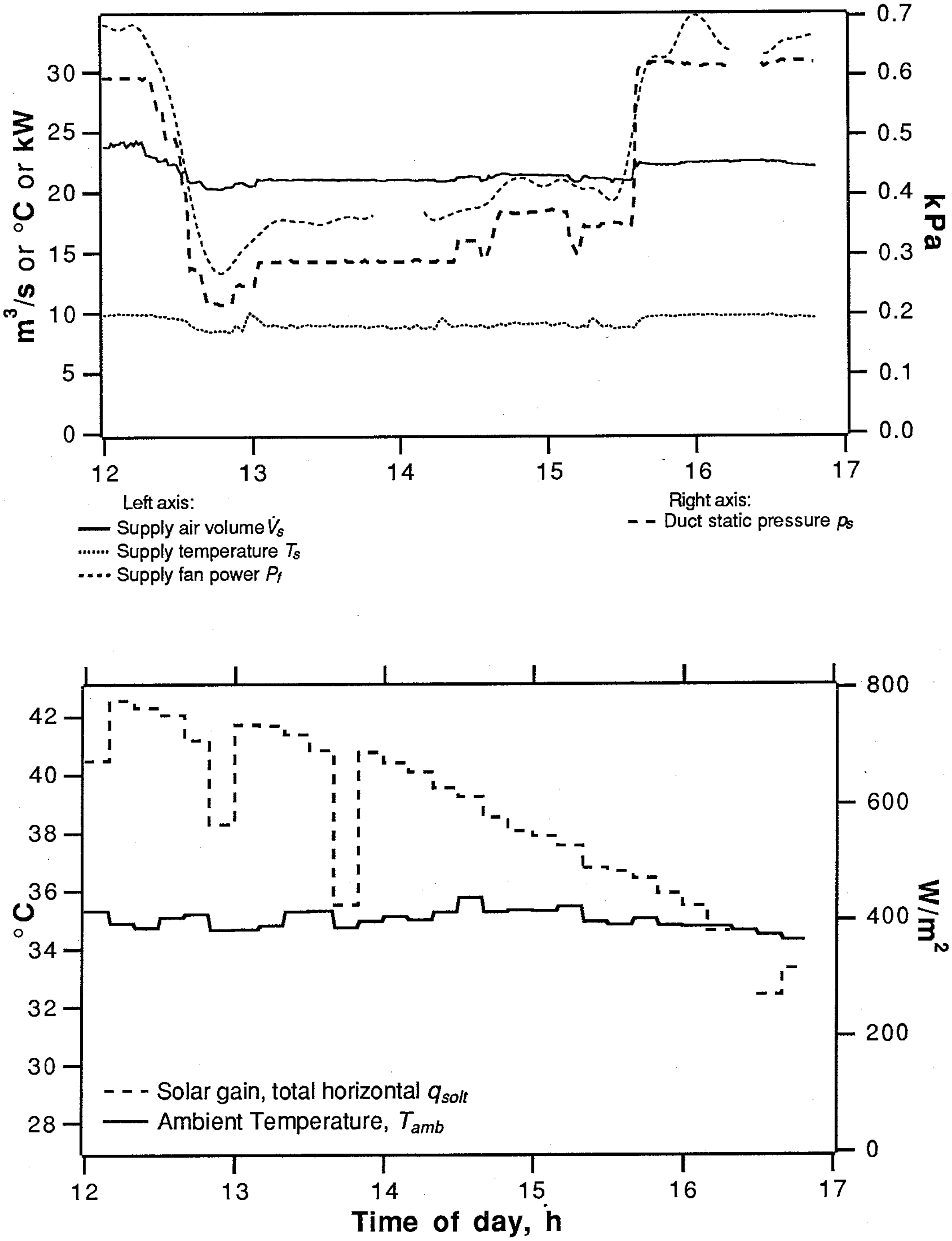

Figure 3.11. Response of supply fan to manual adjustment of static pressure set point; external gain factors (Experiment 1). Supply airflow is relatively insensitive to changes in static pressure; the small changes show the combined effect of the non-linear flow characteristic of pneumatic terminal boxes, as well as those boxes that are either fully open or at minimum flow. Supply fan power follows pressure. The power data time series shown here has been smoothed to eliminate noise caused by poor resolution.

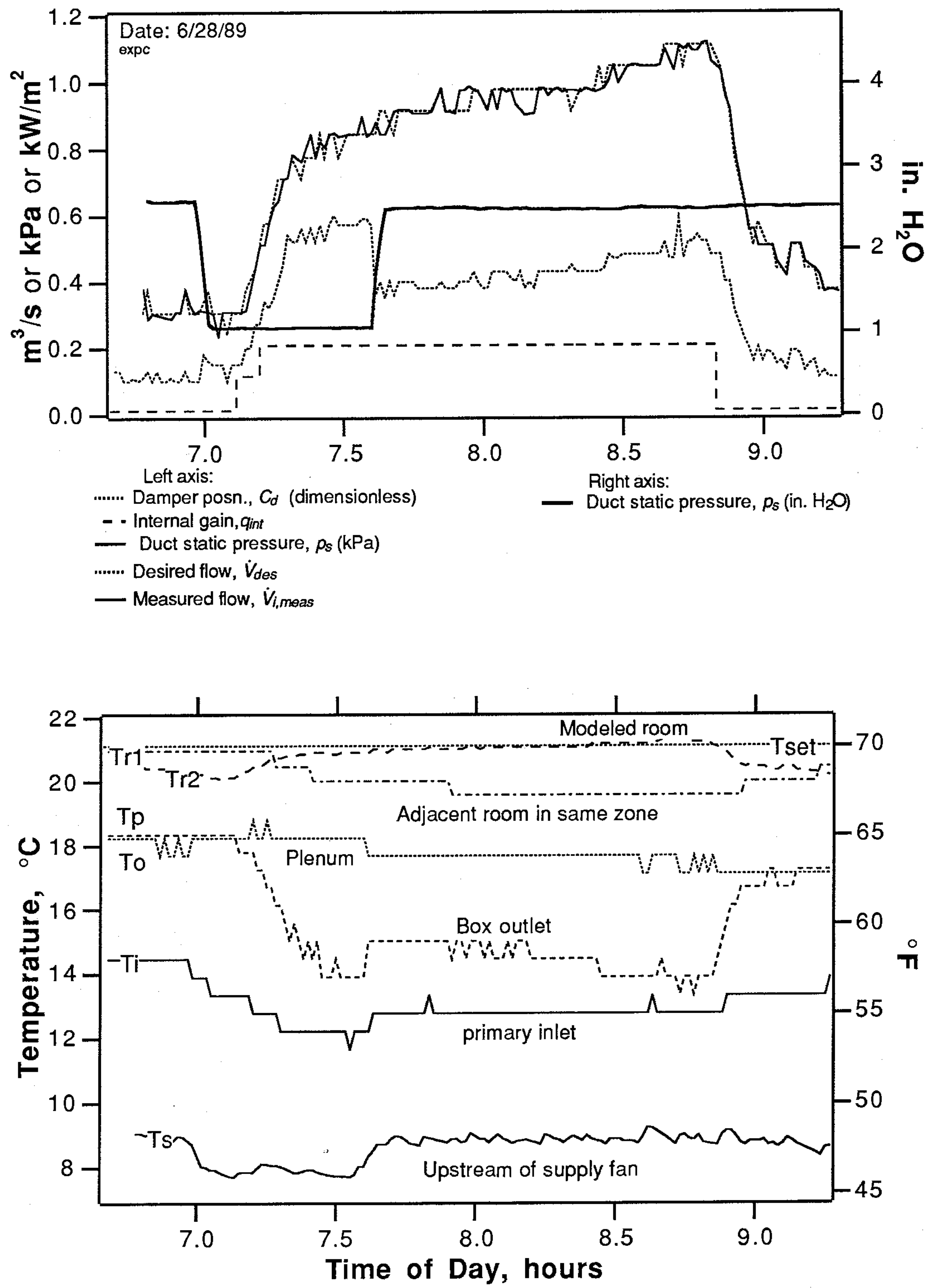

Figure 3.12. Measured response of terminal box and zone to changes in static pressure set point and internal gain (Experiment 2).

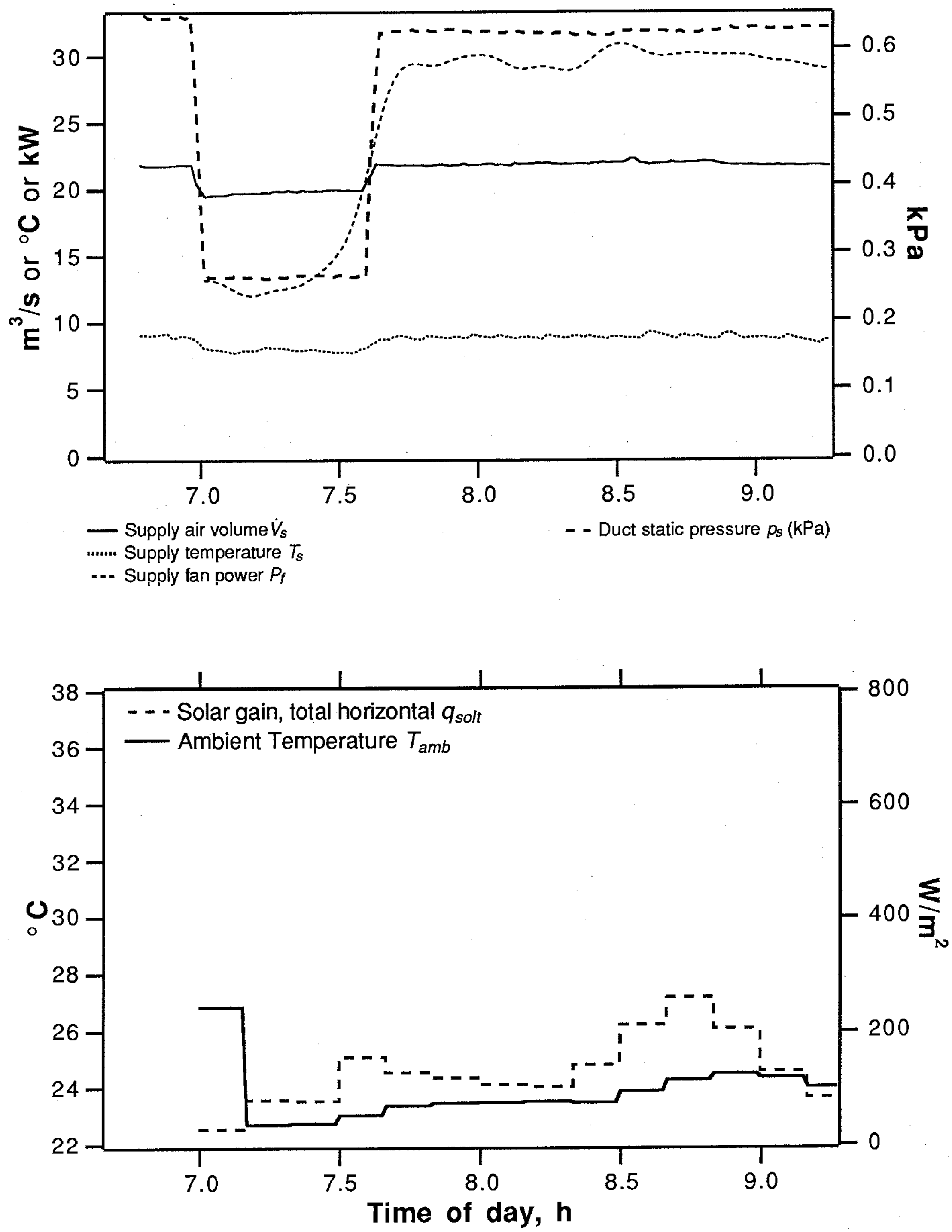

Figure 3.13. Response of supply fan to changes in static pressure set point; external gain factors (Experiment 2).

Before further discussion of the results, it should be noted that throughout all the experiments, plenum temperatures were unusually low, for two reasons: 1) the surrounding zones that exhausted into the same ceiling plenum space were unoccupied and hence lower in temperature, and 2) it was discovered later that another terminal box sharing the plenum had failed open, dumping cold supply air into the plenum (Box "G" in **Figure 3.7**). If plenum temperatures were closer to those of the zone, as they typically would be, the steady state error (due to proportional-only control) in the zone temperature would be higher under similar load and flow parameter (reset range) conditions, resulting in operation of the box closer to its maximum flow rate. Of course, the actual flow rates observed in a given system would depend on both the flow rate limits and the load on the space. No attempt was made here to use "typical" values of these parameters, beyond choosing a reasonable maximum flow rate and loads that would exercise the box through its flow range given the supply air and plenum air temperatures present during the time of the experiment.

Figure 3.10 shows the damper varying between about 25% open ($C_d = 0.25$) and full open ($C_d = 1.0$), in response to changes in $\dot{V}_{des}$ and p_s. The actual flow rate tracks the desired flow rate very well, as the damper response is much faster than the changes in p_s. During this experiment, the box is always able to satisfy the desired flow rate, despite p_s dropping below 1 in. H_2O. Given the introduced heat gains and controller parameters used, the desired flow rate reaches about 95% of maximum.

The steady-state error in room temperature due to proportional-only control is evident especially during the first two hours of Experiment 1 (**Figure 3.10**). The room starts out in equilibrium, with the temperature exceeding set point by 1.5 F, and the flow rate at 50% of maximum. When a load of 1650 W is added at about 45 minutes, the temperature rises to 2.0 F above set point, and flow is two-thirds of maximum. This condition persists for over an hour, with only very slight indication of a decrease in temperature.

The response of the room and terminal box to step changes in load is relatively fast, taking about 10 min. to come to equilibrium. The response takes an exponential shape typical of a simple first order system.

Figures 3.11 and **3.13** show supply air volume varying only slightly to large changes in p_s. The decrease observed during the first hour of Experiment 1 reflects boxes that are either fully open or at minimum flow, and possibly the non-linear flow vs. temperature characteristic of the pneumatic terminal boxes (discussed earlier in this chapter).[15] The response of supply flow to changes in static pressure during these and two other experiments was discussed in Chapter 2, and shown in **Figure 2.18**. Supply fan power follows static pressure relatively closely, i.e., changes in static pressure are accompanied by changes in power of the same proportion, reinforcing (in time series) the relationship we saw in Chapter 2.

Experiment 2 (unlike Experiment 1) begins with a positive steady-state error in room temperature, as there is no internal gain, and the thermal mass of the room is probably still cool from the evening. The addition of internal heat brings the temperature almost to set point within 10 minutes, and causes a steady, but small gain for the next hour and a half until the load is removed. Duct static pressure has essentially no effect on the zone in this case; the damper controller keeps measured flow rate within 0.1 m^3/s of desired, irrespective of duct static pressure. As in Experiment 1, the air temperature upstream of the supply fan is seen to dip slightly in response to the decrease in flow rate across the cooling coil.

[15] i.e., a decrease in static pressure should result in an initial decrease in flow for pneumatic boxes within their throttling range. This effect would disappear if the zone heats up in response to decreased flow, however that would cause a gradual increase in total supply flow, not observed here in Experiment 2.

3.3 TERMINAL BOX AND ZONE SIMULATION USING *HVACSIM+*

Simulation as a tool in buildings research is used in several ways: to estimate the effects on comfort and energy use of various system features during the design phase of a new building or retrofit, to pinpoint causes of energy and comfort problems in existing buildings through comparison with measured data, and to explore design options as part of the manufacturing design process for system components. In many cases, dynamic performance analysis of individual system components is not required, and a simulation tool such as DOE-2, with a calculation time step of one hour, can capture building thermal transients. Evaluating the dynamic performance of control systems requires a tool designed to solve non-linear equations on a smaller time scale, as well as detailed component models.

The primary purposes of simulation in this study are three-fold:

- to explore the potential problems and benefits of several strategies proposed here to improve VAV system control
- to develop a simple model of a constant volume terminal box
- to assess what is entailed in the development and simulation of simple system models using available simulation software

Dynamic simulation was required, and the program *HVACSIM+* (Park et al., 1985), developed at the U.S. National Institute of Standards and Technology (formally National Bureau of Standards), was chosen for several reasons:

- the component models that come with it are experimentally verified and would serve with little modification as the building blocks of the systems modeled in this study
- the modular architecture of the program allows component models to added or modified easily
- the software is in the public domain, enabling other researchers to easily make use of and verify the models developed in this study

HVACSIM+ was chosen over a similar (and more widely used) program, *TRNSYS* (Klein et al, 1976), because the techniques it uses are thought by some to make it better suited for control problems. While many aspects of the design of *HVACSIM+* were borrowed from

TRNSYS , it differs in several respects, employing a central equation solver that solves stiff non-linear differential equations simultaneously to obtain a self-consistent solution at each time-step (Hill, 1985). Integration is performed using a variable time step, variable order algorithm, enabling the separation of subsystems with radically different time scales. In addition, the components in the library distributed with *TRNSYS* do not model pressure of the fluid.[16]

All of the simulations in this study were performed with *HVACSIM+*, either running on a DEC VAX-8700 under the ULTRIX operating system, or on a SUN 3/60 under SunOs UNIX. The Fortran source code was compiled on each machine using the f77 Fortran-77 compiler.

The system models developed in this study were designed based on measured characteristics of the experimental system. While attempts were made to calibrate the performance of individual component models and systems to their actual counterparts in order to choose physically realistic parameters, a more rigorous calibration was unnecessary, and would have required a considerably greater investment in time, as well as more comprehensive measurements of the test equipment characteristics. The system models developed here were designed mainly to provide a first look at alternative VAV control strategies.

The terminal box/zone system was broken up into three "blocks" for simulation purposes: the terminal box, the duct between the box and the rooms, and the zone (**Figures 3.14, 3.15**, and **3.16**). Only the room in which the thermostat was located was modeled. This division into blocks was done in order to reduce computation time: the time required to solve a set of equations (i.e., a block) is proportional to the square of the number of equations. *HVACSIM+* solves all equations in a block simultaneously (Clark and May, 1985). All equations in a "superblock," a grouping of blocks, are also solved simultaneously, based on preliminary solution of the block equations. Superblocks are

[16]They could be modified, however, to model pressure, which would involve adding the appropriate equations to the source code for each component.

solved sequentially, and are hence only loosely coupled. They may be run with different time steps. Here, the terminal box and the duct make up one superblock; the room is in its own superblock with larger time steps than the other, as its response time is much longer.

Each block is constructed by connecting components selected from a library. The only component that was not already in the library was the flow sensor, created easily by modifying the temperature sensor model. The physical parameters for each component were estimated from field observations, mechanical drawings, and manufacturers' specifications. Flow resistance coefficients were calculated based on tables (following Chapter 32) in ASHRAE (1989), and then adjusted by simulation under time-invariant boundary conditions to match measured flow rates and pressure drops.

The *HVACSIM+* component models relate pressure losses to flow rates using equations of the following form (Clark, 1985):

$$\Delta p = K\dot{m}^2 \tag{3.3.1}$$

where

Δp	=	pressure drop across the component
K	=	flow resistance or pressure loss coefficient
$\dot{m}$	=	mass flow rate

Simulations must be set up such that the set of equations is neither under-determined nor overdetermined. Information about how each component handles pressure and flow calculations is required to do this. Components can be categorized into four groups on this basis:

- components with no fluid flow (e.g. the controllers and sensors in this model)
- components with fluid flow but no pressure drop (e.g. the room)
- components in which inlet pressures are calculated from outlet pressures and inlet flow rates (e.g. the fan, conduit, merge and split)
- those in which flow rates are calculated from inlet and outlet pressures (e.g. inlet conduit)

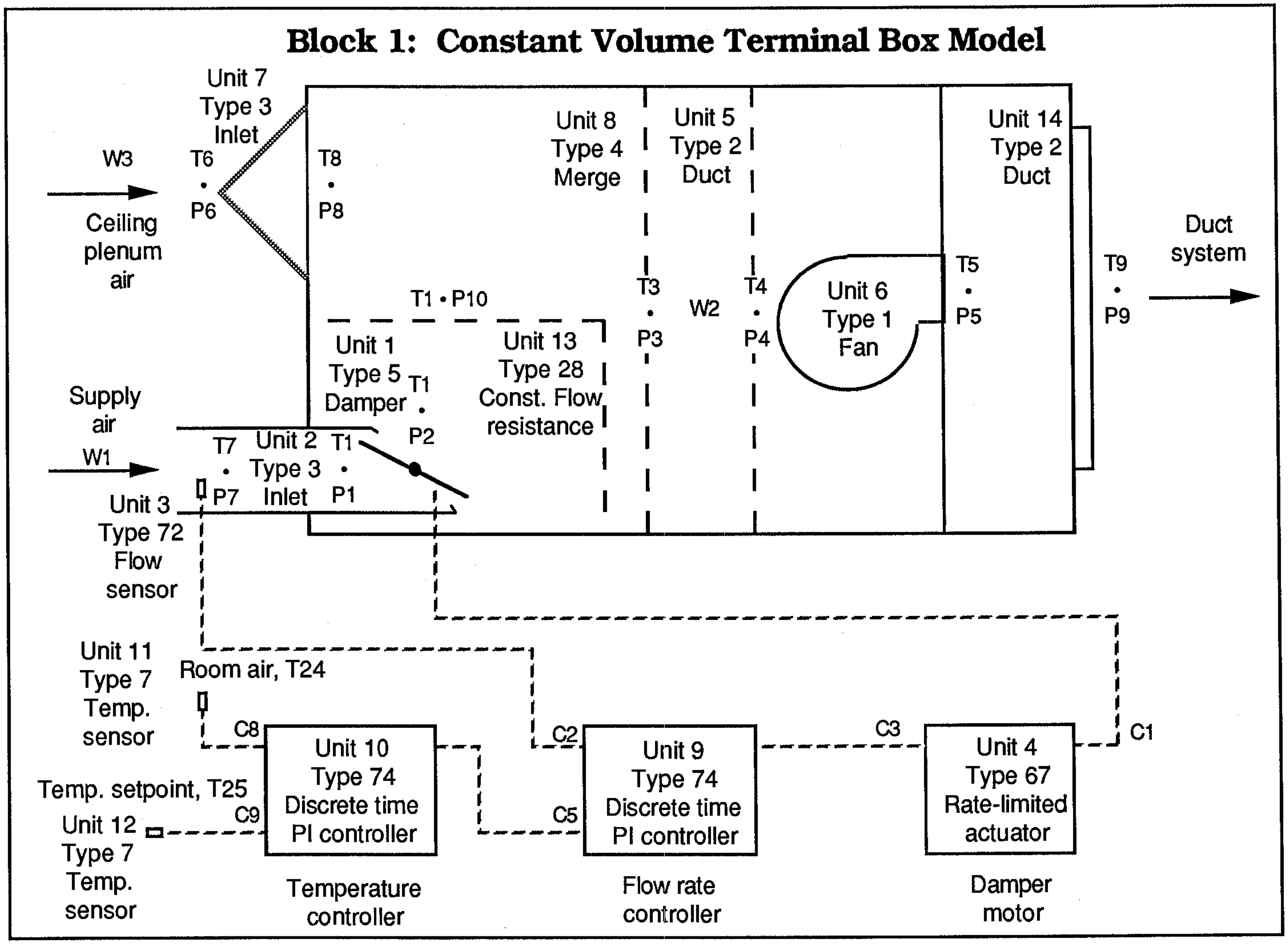

Figure 3.14. *Terminal box model for use with HVACSIM+. Types are components in the HVACSIM+ library. The flow sensor (type 72) was developed in this study; the controllers and actuator were modified from models developed by others.*

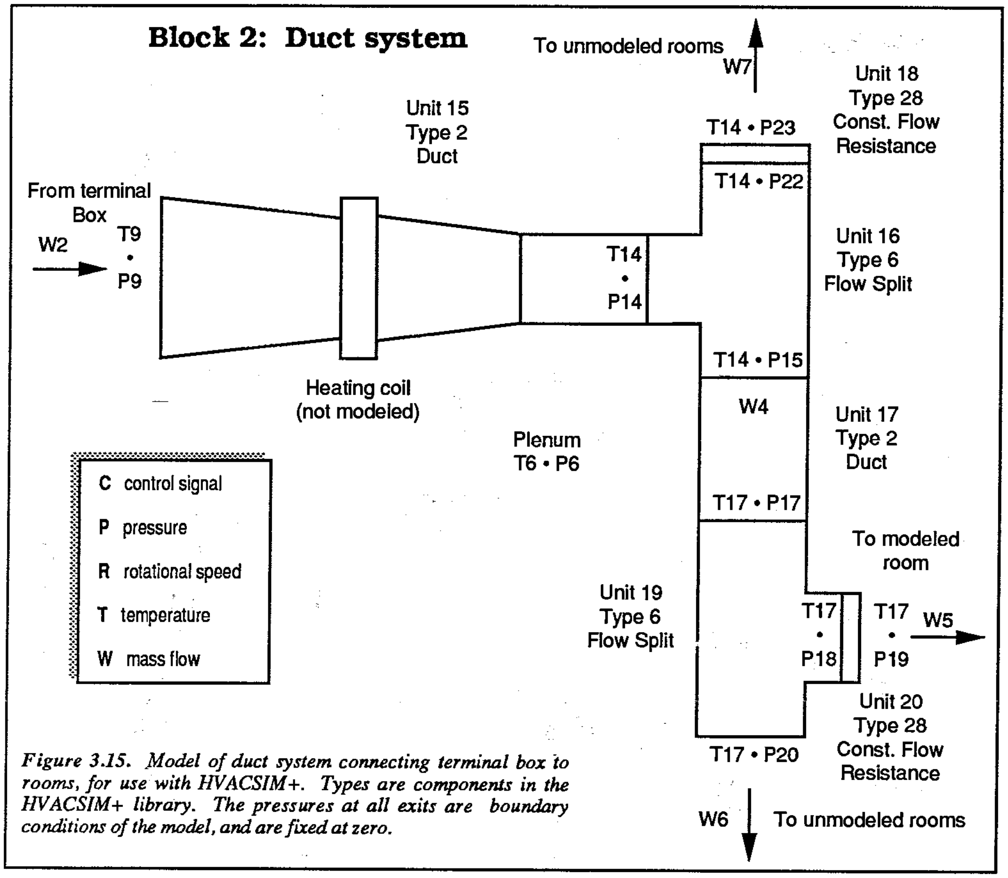

Figure 3.15. Model of duct system connecting terminal box to rooms, for use with HVACSIM+. Types are components in the HVACSIM+ library. The pressures at all exits are boundary conditions of the model, and are fixed at zero.

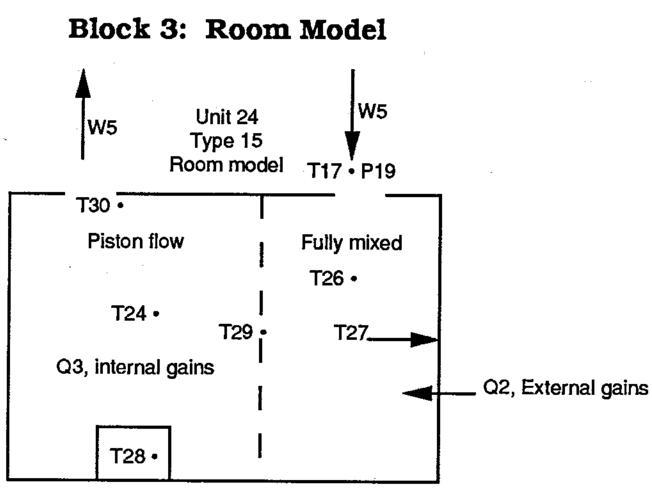

Figure 3.16. HVACSIM+ room component model. External gains enter the wall mass rather than the room air directly. The air mass is divided into two parts: a well-mixed portion near the supply, and a piston-flow portion near the exhaust.

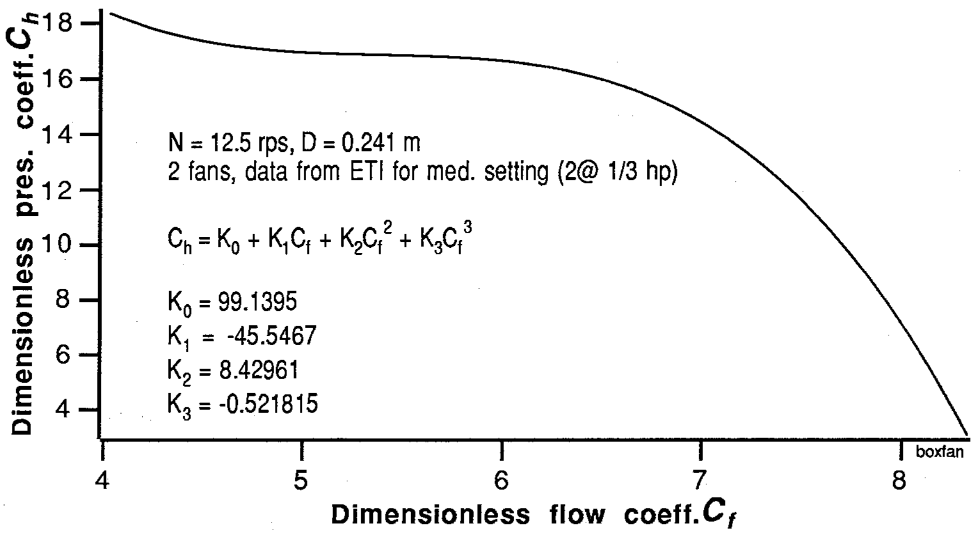

Figure 3.17. *Determination of pressure coefficients for dimensionless fan curve using regression of manufacturers' data (Appendix B) for 14" terminal box fan. Coefficients serve as parameter inputs for the* HVACSIM+ *fan component model.*

In general, including one component of the fourth category in each flow circuit will enable the determination of a flow rate and hence a unique solution. The inlet pressure of this component serves the boundary condition from which all other pressures in the system are determined. In the model presented here, the ceiling plenum inlet of the terminal box serves this function.

THE COMPONENT MODELS

Fan. The parameters for the terminal box fan were determined from a manufacturer's fan curve (Appendix B and **Figure 3.17**). The speed of the fan was adjusted in the simulation to give the desired flow rate for a given pressure rise across the fan. Although efficiency is modeled as a polynomial function of flow, this was set equal to a constant, since the fan would be operating at a more or less constant volume. Using measured temperature rise across the fan, airflow and fan electrical power, the efficiency was adjusted to 0.2 in order to match this characteristic. While this is not a realistic value, it is of little consequence here, as terminal box fan energy use was not a consideration in the simulation.

Damper. The damper is a key part of the terminal box model. The flow resistance coefficient for a given damper varies with position using the following *HVACSIM+* model:

$$K = \frac{F_w K_{wo}}{\left[(1-\lambda)C_d + \lambda\right]^2} + (1 - F_w)K_{wo}\lambda^{(2C_d - 2)} \tag{3.3.2}$$

where

F_w	=	weighting factor for linear term of flow resistance coefficient
K_{wo}	=	flow resistance coefficient, wide open
λ	=	leakage parameter: fractional leakage when $\Delta p = 1$ kPa
C_d	=	relative damper position

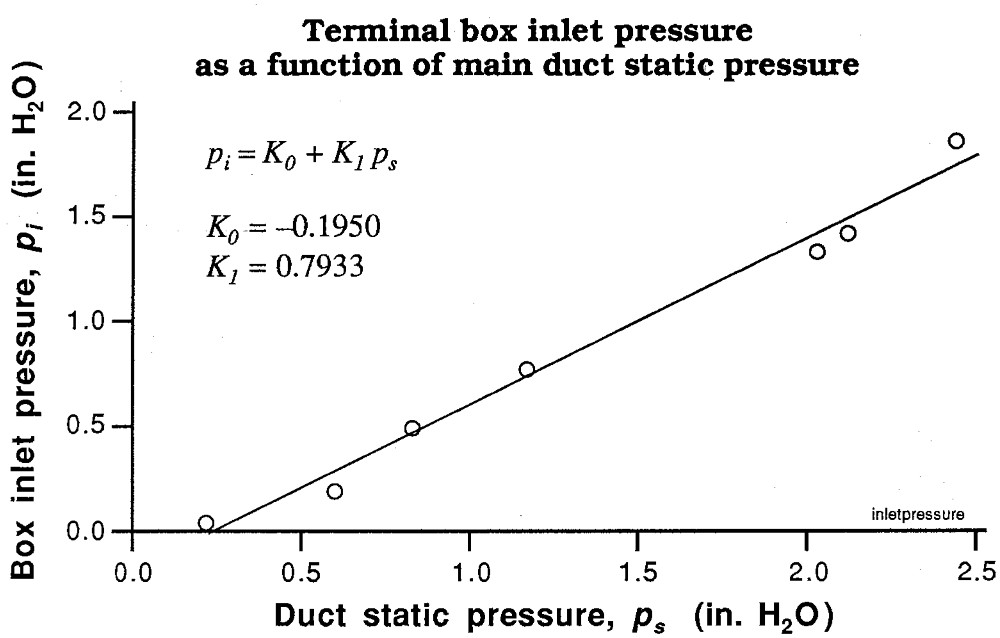

Figure 3.18. Measurements of static pressure at the terminal box inlet (on the first floor), taken with an EDM digital micromanometer, plotted against values of static pressure at the main system sensor (on the third floor) measured using a pressure transducer connected to the data acquisition system. The non-zero intercept is approximately equal to the difference in static pressure due to elevation, as the main sensor is about 7.6 meters higher, and each pressure measurement is relative to local room pressure.

Experimental measurements of inlet pressure, flow, and damper position were used to determine the parameters. The pressure drop across the damper was not measured directly, however when the primary air flow is at a maximum (i.e. equal to the total flow through the box), the pressure downstream of the damper is approximately zero, hence the pressure drop is equal to the inlet pressure. The box was not instrumented with an inlet pressure sensor, but manual measurements of inlet pressure correlated well with automatic measurements of duct static pressure at the main building sensor (**Figure 3.18**), so inlet pressure can be estimated using the regression coefficients.

K is then calculated as the inlet pressure divided by the mass flow rate squared, for cases when primary air flow is equal to the total air flow through the box. **Figure 3.19** shows these data plotted against damper position. The parameters were determined by fitting a curve of the form of equation 3.3.1 through the calculated values of K at each point, using non-linear regression. Two regressions were done, one with all three parameters free, and another with only F_w free, and λ and K_{wo} constrained to estimated values. Several

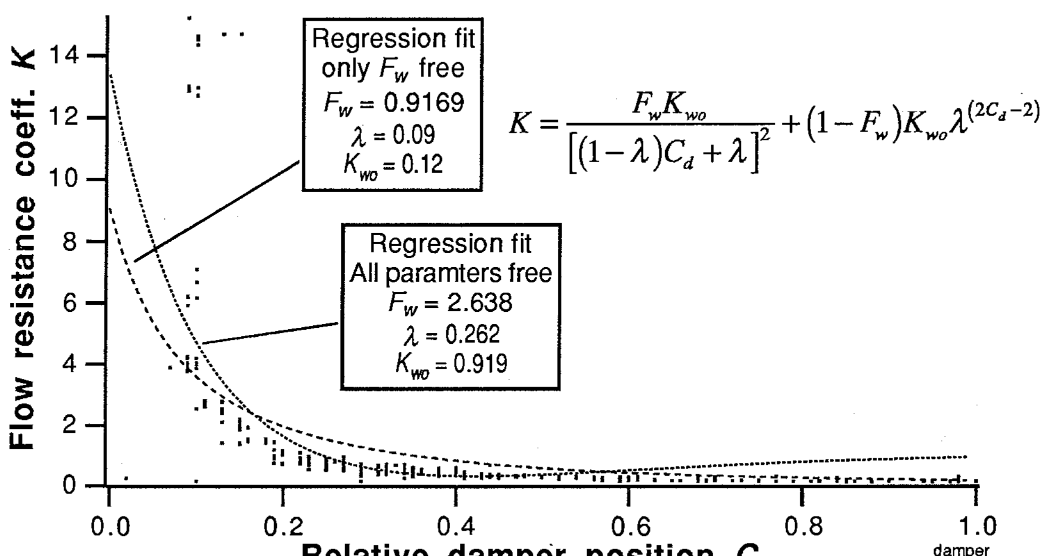

Figure 3.19. Determination of flow resistance parameters for damper model. The leakage and full-open resistance parameters were determined using iterative simulation and regression. The free-fit parameters were not used, as they were not physically reasonable, and did not represent the damper well at positions near full-open. It appears that the measured performance of this damper differs significantly from the model at positions below 10% open, where the resistance coefficient rises sharply.

measurements of pressure and flow with the damper fully open and primary flow at maximum yielded a relatively consistent estimate of K_{wo} as about 0.153, however only one (poor) estimate of the leakage factor λ, 0.13, was obtained. The parameters determined using the unconstrained regression were not used because they were not reasonable from a physical standpoint. The constrained regression resulted in a F_w of 0.876. These values were tested in the simulation; the leakage parameter was found to be too high and was adjusted to 0.09, and K_{wo} was adjusted to 0.12. These values and the F_w of 0.917 resulting from subsequent regression yielded good simulation test results. The fact that F_w is close to 1.0 indicates that the flow controlling action of the damper is fairly linear (above $C_d \approx 0.1$).

It is clear from the discrepancy between the data and the fit to the model that an equation of this form does not represent the behavior of the damper well for positions of less than 10% of full open. This is probably because the damper used here is closed at 45° from full open, while the dampers on which the model is based have a span of 90°.

Ducts. The allocation of flow in the model between this room and the unmodeled spaces was done using the following procedure:[17]

1. Initial estimates of resistance coefficients for the duct components were made using tables following Chapter 32 in ASHRAE (1989). Since *HVACSIM+* models flow splits (wyes) as having the same pressure drop through each branch, the flow resistance for these components was set at the lower of those for the two branches, and a constant resistance for the balance was added downstream of the branch with the higher resistance. The resistance coefficients published in the above reference for various duct configurations are given in the form

$$C_o = \frac{\Delta p}{\rho v^2} \tag{3.3.3}$$

and must be converted for use in *HVACSIM+* using the relation

$$K = \frac{0.001 C_o}{\rho A^2} \tag{3.3.4}$$

[17]Although total air flow to the zone was measured experimentally, the portion going to the modeled room was not. Duct runs downstream of disturbances were too short, and measurements at diffusers would have required a flow hood, which was not available. The estimation procedure described, however, provided suitable values.

where A is the cross-sectional area and v is the velocity. The factor of 0.001 must be included for use with pressures in kPa.

2. A measurement of pressure upstream of the heating coil was made when the total flow through the box was equal to that shown on the air balancer's report (Knice, 1985) for this zone.

3. A simulation of the duct model was run with this pressure as the inlet boundary condition, and a pressure of zero at the outlets. Using the outlet flow rates in the balancer's report as a target, the resistance coefficients of the components were adjusted until the flow through each branch differed from the target rate by the same percentage.

4. The resistance coefficients were increased or decreased in inverse proportion to the square of the flow discrepancy, bringing the flow rates within 2% of the target rates. Individual resistances were then adjusted once more for fine tuning.

5. The pressure at the duct inlet was increased to that measured under the higher flow rates used during most of the experimentation, as a check on the model. The resulting total flow through the duct system was within 1% of the measured value, confirming the resistance parameter selection and enabling estimation of the flow rate into the modeled room during the experiments.

Controllers and actuator. While the continuous time PI controller included in the *HVACSIM+* library would be appropriate for modeling the pneumatic terminal box temperature and flow rate controllers, a discrete time model is more representative of the controller used in a DDC terminal box.[18] A model developed by Clarke (1983) and implemented by Haves (1985) was used here, after modification to include a deadband and input limiting.[19] The algorithm is similar to that described by Stoecker and Stoecker (1989, pp.410-411), except that trapezoidal rather than backward differencing is used to integrate the error signal. This involves using the value of the error at the current rather than previous time step in the integration, resulting in greater accuracy and stability when the integral

---

[18]Another advantage of using a discrete-time controller model here is that it is less computationally intensive than its continuous time counterpart, having only one differential equation (for the transient effect) instead of two (one for the transient, one for the derivative of the integral term). Also, the basic *HVACSIM+* library component (continuous-time PI controller) has no output limiting; using two controllers in series is necessary if this is required. Thus only one differential equation is required, instead of four, although it would be easy to modify the continuous controller to include output limiting.

[19]Limiting, or clipping to the minimum or maximum of the range (0 and 1 here) input signals that fall outside the range is representative of what occurs in a real device. *HVACSIM+* models typically limit input, rather than output signals.

time approaches the sampling time. The algorithm is also formulated in a manner that inhibits integral wind-up:

$$C^{ss}_{k+1} = K_P E_{k+1} + I_k$$

$$I_k = \beta I_{k-1} + (1-\beta) C^{ss}_k$$

$$\beta = \frac{(2T_I - \Delta t)}{(2T_I + \Delta t)} \qquad (3.3.5)$$

where

C^{ss}_{k+1} = steady-state output at time $k + 1$

K_P = proportional gain

β = $1 - K_I^*$, where K_I^* is the effective integral gain = $\Delta t / T_I = \Delta t K_I / K_P$

Δt = controller sampling interval

This can be also expressed in Stoecker's incremental controller form as

$$\Delta U_{k+1} = K_P\left[E(k+1) - E(k) + K_I^* E(k+1)\right] \qquad (3.3.5b)$$

where ΔU_{k+1} is the incremental controller output at time $k + 1$.[20] The transient response of this model is identical to that of the continuous-time model.

The deadband is implemented here by calculating an effective error, E' from the actual error E and the deadband d (Stoecker and Stoecker, 1989):[21]

$$\begin{array}{ll} -d \le E \le +d: & E' = 0 \\ E > d: & E' = E - d \\ E < -d: & E' = E + d \end{array} \qquad (3.3.6)$$

[20]The incremental form, shown here for reference, is not used in the controller model for reasons of convenience: the Clarke/Haves code, with modifications, was suitable, and the rate limited actuator model required a position input, rather than an incremental one.

[21]The "slow band" described for the actual damper controller in Section 3.1 was not included in the model.

Source code for the controller is given in Listing C.1 in Appendix C. A sampling time of 10 s and a deadband of 1/256 of full scale or 0.00391 were used to represent the DDC controller used in the experiment.

The output of the flow rate controller serves as input to a rate-limited actuator model, simulating the synchronous motor of the DDC terminal box. The actuator model, also by Haves (and Dexter, 1989), was modified to conform to the linkage geometry of the damper actuator used in the experiments. A limit-to-limit time of 180 s was specified. The gain parameters of this controller were tuned by trial and error using simulated response of the system to a step change in commanded flow rate. After selecting a suitable proportional gain,[22] with the integral gain set at 0, the integral gain was adjusted so as to minimize steady state error in the measured flow rate (**Figure 3.20**).

As in Equation 3.1.9, the integral gain of the temperature controller is zero, the constant integral portion of the output is equal to the zero offset, and the proportional gain is

$$K_P = -\frac{T_g}{R_T} \tag{3.3.7}$$

Sensors. Temperature and flow sensor models are identical with the exception that each requires a different type of input signal. The flow sensor is modeled as follows:

$$C_i = \frac{\dot{m}_i - \dot{m}_o}{\dot{m}_g}$$

$$\frac{dC_o}{dt} = \frac{C_i - C_o}{\tau} \tag{3.3.8}$$

where

C_i = modified input signal

C_o = output signal

$\dot{m}_i$ = flow input (kg/s)

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[22]Since the slow actuator dominated the transient response, this parameter was not critical.

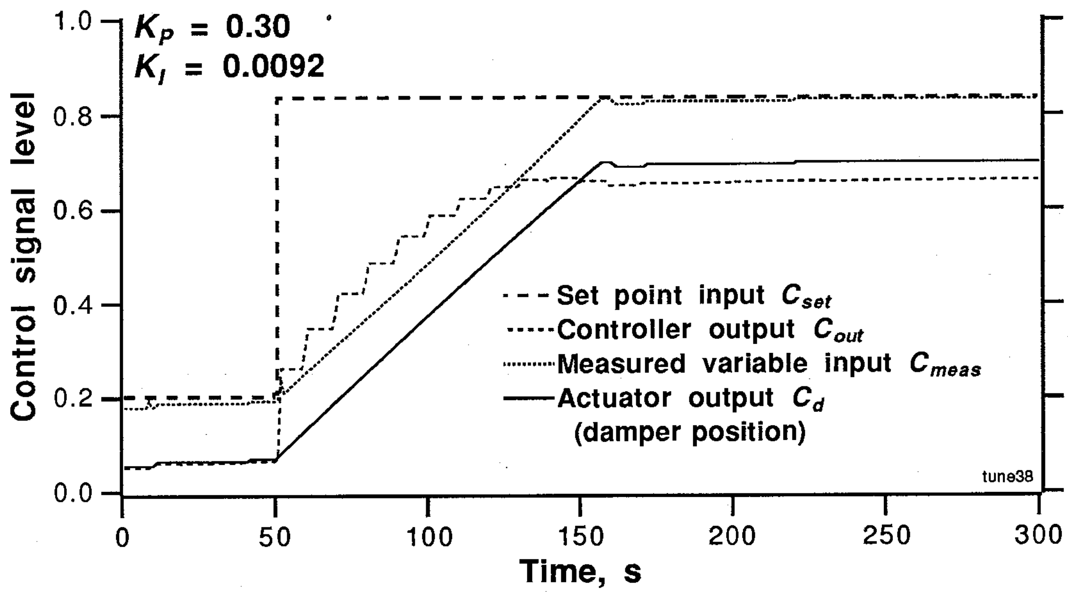

Figure 3.20. Gain parameters of the damper controller were chosen by trial and error. Final choice of proportional and integral gains are shown. The measured variable is flow rate; the controller's output serves as input to a fixed rate actuator that controls damper position. Controller sampling interval is 10 s, actuator travel time is 180 s, limit to limit. The controller output changes at each sampling interval, as is typical of discrete time controllers. The fixed rate actuator has a linear response.

$\dot{m}_g$ = gain = $\rho \dot{V}_{up}$

$\dot{V}_{up}$ = nominal upper limit of flow rate, taken as 1.42 m^3/s (3000 cfm)

$\dot{m}_o$ = offset = 0.0

τ = time constant, taken as 0.5 s[23]

A temperature sensor is used as the thermostat set point so that this input may be expressed in the same units as room temperature (°C). The equivalent parameters for the temperature sensor and set point are

T_o = lower range of temperature sensor, taken here as 13.33 °C

[23]No good estimate of this parameter was available. The value chosen is probably fast enough that an instantaneous device would not have produced significantly different results for the time scale of interest. This did have the advantage, however, of smoothing out occasional numerical discontinuities in the calculated flow rate due to non-convergence.

T_g = upper range of temperature sensor, taken here as 18.89 °C

The time constant of the temperature sensor was taken as 60 s.

Room model. The *HVACSIM+* library contains sophisticated component models for simulating zones or whole buildings under typical or actual weather conditions: zone envelope, surface construct, zone model, and weather input (Park et al., 1986). Modeling the zone itself to this level of detail, however, was not the purpose of this study, so a simpler room model component was used here. This model, based on work by Borresen (1981), and described in Hill (1985) has far fewer parameters, making it easier to use, and treats internal and external heat gains as inputs. The air mass is divided into two parts: a well-mixed portion near the supply, and a piston-flow portion near the exhaust. The room mass is divided into an interior mass and a wall mass, which each exchange heat with the two air masses. Internal gains are represented by a heat flow into the mixed air mass[24], and external gains (losses) are represented by heat flow into the wall mass (Clark, 1985).

The room model was implemented in these simulations by considering the mass of the insulated floor slab and interior walls as the interior mass and the inside layer of gypsum wall board and insulation in the exterior wall as the wall mass. The heat transfer coefficients and material properties were estimated based on information found in ASHRAE (1989, Chapter 22, Table 4), and Holman (1986, Table A-3).

Experimental Verification and Calibration of the Terminal Box/Room Model

In order that the room model would respond to inputs in a manner similar to that of the experimental room, the room component alone was simulated using values of ventilation air temperature and internal gain estimated from experimental measurements as bound-

[24]In a real room, much of the internal heat gain is actually radiated directly to the mass.

ary conditions.[25] The heat transfer coefficients, masses, initial mass temperatures, and external gain were adjusted until the room temperature (piston flow portion) response was reasonably similar to the observed response of the experimental room. It was found that an external gain function of the form

$$q_{ext} = C_1 \cdot \exp(-t / C_2) + C_3 \tag{3.3.9}$$

where q_{ext} is in kW, t is the time in seconds from the start of the simulation, and C_x are constants, provides satisfactory results. Since in this implementation of the model, external gain is considered to be the heat gain to the interior section of the exterior wall (from the higher temperature exterior stone section) plus conduction through the glazing[26], an equation of this form is reasonable, the decaying term representative of the decay in temperature of the exterior stone section of the wall previously warmed by the sun. After several trials, the constants were taken as $C_1 = 2.5$, $C_2 = 3600$, and $C_3 = 0$.

The resulting temperature response in shown in **Figure 3.21**,[27] compared to the measured temperature. Although the fit is poor toward the end of the simulation, differing by almost 1 °C from measured, this was acceptable within the limited scope of this study. Further adjustment of the parameters and initial values and perhaps reallocating mass between the wall and internal masses would probably improve the results. This procedure would be trial and error, however, and would be time consuming, considering the complexity of the situation with so many unknowns, even for such a simple model. It is possible that the limitations of the model, such as the lack of distinction between wall and floor mass (which may also exchange heat with the surroundings), may prove that the more complex building components of *HVACSIM+* are better suited for analysis such as this in

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[25]Since ventilation air inlet temperature was not measured, this was estimated using measured terminal box outlet temperature and a representative value of duct heat gain observed in simulations of the whole zone system. The result was smoothed (ten passes) so as to be suitable as input for *HVACSIM+*.
[26]There was no direct solar gain during the period during which the experimental data used in this calibration were collected.
[27]Note that the measured temperature response is smoother (neglecting sensor noise), probably due to the effect of the sensor's time constant.

the absence of measured room component temperature histories. The discrepancies do not necessarily diminish the usefulness of the simple room model for assessing terminal box performance, however, and any improvement offered by the more complex one may not be worth the added effort in computation and parameter selection.

In order to validate the system model, modified[28] measurements of p_s, T_i, T_p, and internal gain from this experiment were used as time-varying boundary conditions for the simulation. The model was first "initialized" for 360 seconds to let the equations converge.[29] The final states of variables in this run then served as the initial conditions for variables that were not specified as time-varying boundary conditions. Listing C.7 in Appendix C gives parameter values for this simulation. The results are shown in **Figures**

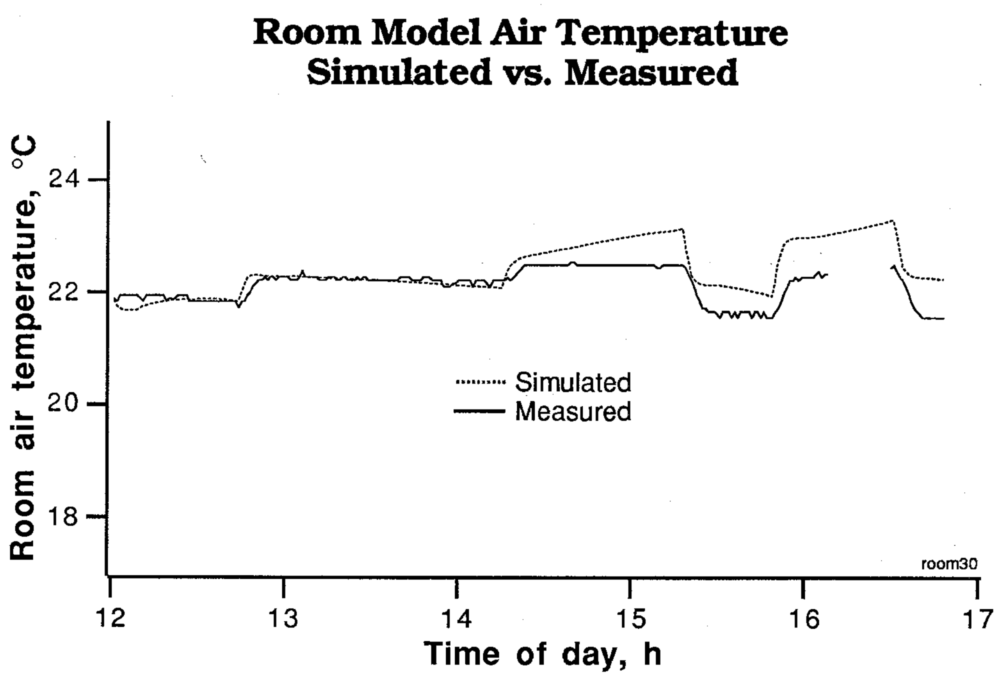

Figure 3.21. The room component only was simulated, with measured ventilation air temperature as a time-varying boundary condition.

[28]Inlet pressure was determined from p_s as described above. Time-varying boundary condition data (except for q_{int}) were smoothed slightly (10 passes) so that the simulation would perform better. This is discussed further in Chapter 5.

[29]I.e., run using the initial values of time-varying boundary conditions as constant boundary conditions. This is recommended by the authors of *HVACSIM+*.

3.22 and **3.23**, and are remarkably similar to the experimental results presented above (**Figure 3.10**). Notable differences are:

- The room temperature (and thus desired flow) differ from levels seen in the experiment, presumably due to discrepancies between the actual and simulated temperature responses of the room model. These discrepancies, as discussed above, can be attributed to the assumptions made about initial conditions and external gain, parameter selection, and limitations of the room model.

- The transient room temperature response of the simulation to large changes in load appears to be slightly faster than the experimental response, as shown in **Figure 3.24**. Note, however, that the experimentally measured temperature includes the effect of the sensor time constant; the variable plotted here is the calculated temperature of the room, not the output of the sensor model, which was not recorded in this simulation.

- The response of actual flow rate $\dot{V}_i$ to changes in desired flow rate $\dot{V}_{des}$ and inlet pressure p_i is smoother than seen experimentally. This is most likely because the damper control method used in the simulation is similar to, but not identical to the actual method.[30]

- Although the damper was able to operate well, given the range of flow rates and inlet static pressures, it did not exhibit the characteristics of the actual damper at low flow rates and damper positions. This follows from the poor fit at small damper angles of the measured resistance versus position data discussed above.

Despite these differences, this simulation showed the terminal box/room model to be valid for the purposes of this study. Clearly, refinements (particularly to the room and damper components) would be necessary before using the model for other simulation purposes such as zone temperature control design.

[30]Discussed under "Controllers and Actuator," above.

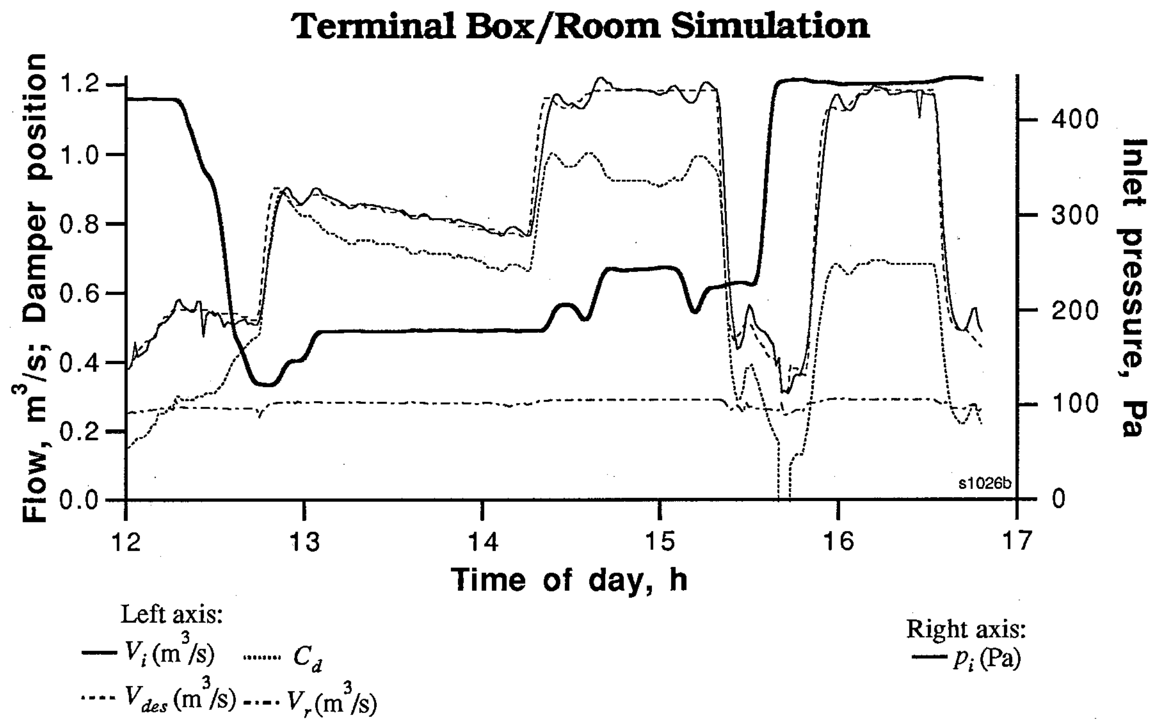

Figure 3.22. Simulated room and terminal box, using measured inlet pressure, inlet temperature, plenum temperature, and internal gain as time-varying boundary conditions.

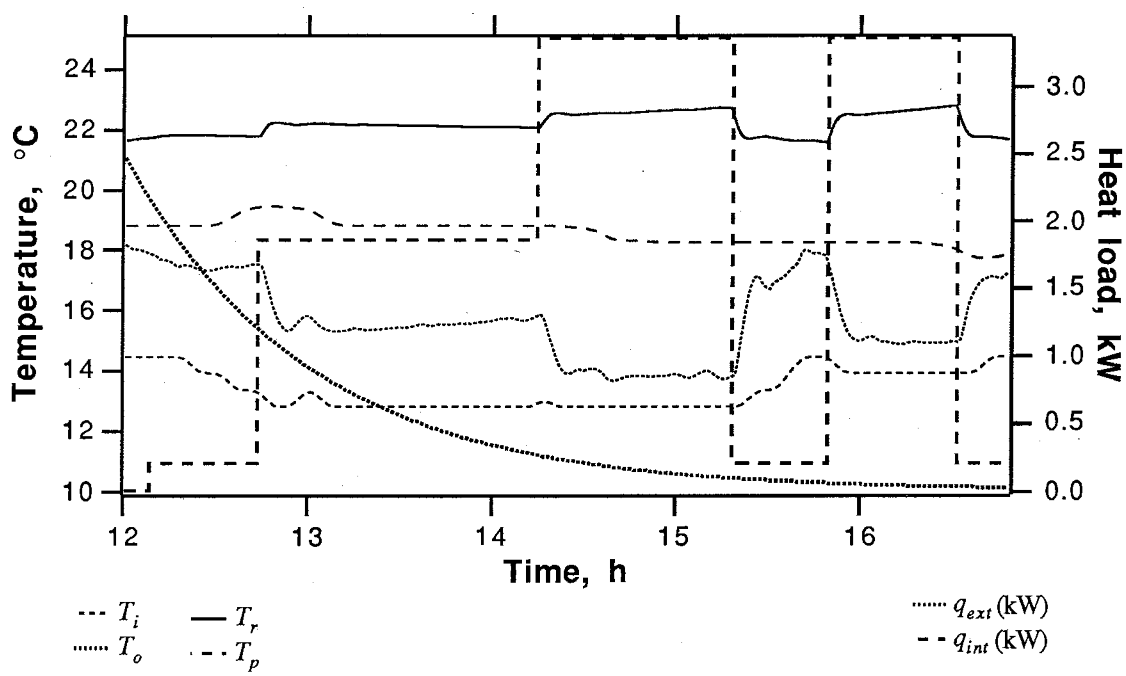

Figure 3.23. Simulation boundary conditions and results, showing internal and external gains, inlet and plenum temperatures, box outlet temperature, and room temperature.

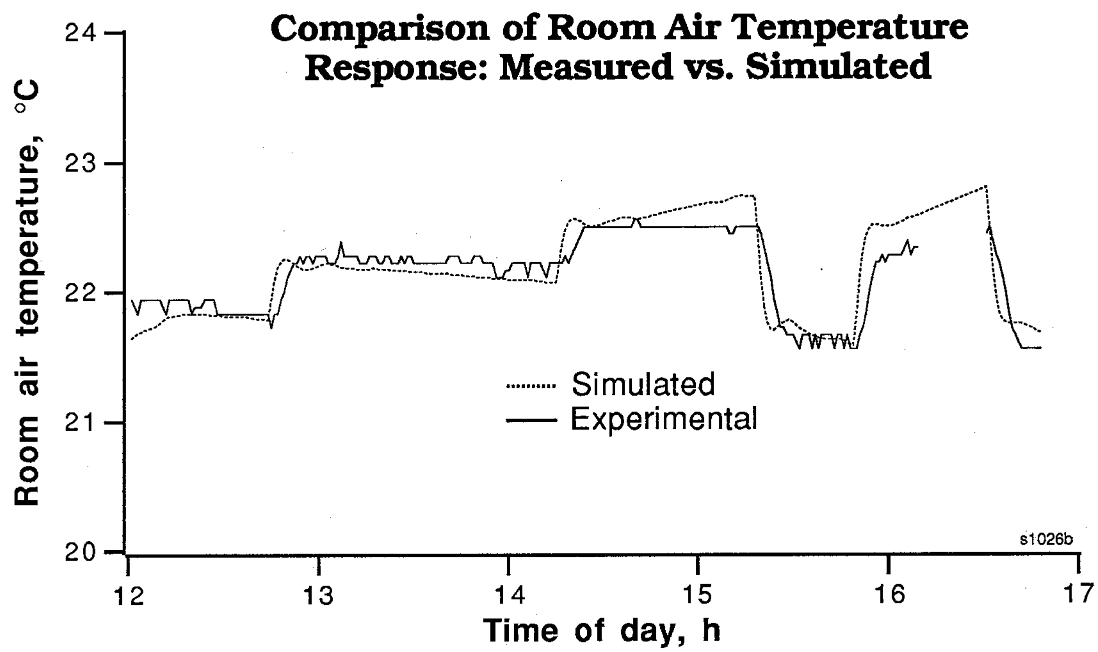

Figure 3.24. Comparison of room air temperature for the simulated system to measured values.

3.4 Two Strategies for Improving Zone Temperature Control

DDC terminal boxes offer flexibility in scheduling operation of zones that is not possible with pneumatic control. Building operators may schedule some zones to be in an "unoccupied" mode (with a reduction in minimum ventilation air, and manual override at the thermostat), while others operate normally. Heating and cooling temperature set points may be controlled remotely and change according to schedule. A late worker may have the capability (at no additional cost in wiring) to put the box into recirculation mode from a switch at the thermostat, even though the supply fan is off. In addition to greater flexibility, DDC terminal box systems can offer the high speed and accuracy that are essential in certain situations. One particular application—control of fume hood face velocity, room pressure, and temperature in laboratories served by VAV systems—offers greater reliability and occupant safety (Marsh, 1988), and is thus a driving force in the development of DDC terminal box technology.

DDC terminal boxes show potential for improving zone temperature control through two strategies in particular: PI zone temperature control, and automatic rebalancing.

PI ZONE TEMPERATURE CONTROL

The experiments and simulations described in the previous section, in which the terminal box was in cooling mode, all exhibit a positive steady-state error in room temperature. The extent of this error is greater when the box is starved and cannot supply the desired primary air flow rate. Although the temperature controller gain was not varied in either the experiments (by decreasing the throttling range) or the simulations, increasing it would have probably decreased the error but not eliminated it under diverse load conditions. A simple and common classical control solution to this problem is to add an integral mode to the proportional controller. Analog electric and proportional PI controllers are available, but have not been used on equipment such as terminal boxes because they cost more to produce than proportional-only controllers (Stoecker and Stoecker, 1989). With DDC control this is no longer a barrier, as PI control (or an equivalent method that controls to set point) can be implemented in software with little added cost.

PI control of zone temperature was not simulated in this study—there is little doubt that it would work; testing it would have been trivial, largely an exercise in controller tuning. At least one manufacturer, in fact, has a DDC terminal box on the market that includes this feature (Anderson, 1989).[31] An issue perhaps more important than technical feasibility is the benefit to comfort and energy efficiency that PI zone temperature control might afford. We have seen that when the terminal box is cooling the space, the steady state error in temperature is negative, i.e., the room is warmer than set point, although it can be positive during a period of low cooling load, as observed in Experiment 2. Although no mea-

[31] Actually, the box uses proportional plus integral plus derivative (PID) room temperature control, but is generally configured with the derivative gain equal to zero.

surements were made with the box in heating mode, the error in this mode would generally be positive, i.e., the room is cooler than set point.[32]

Would bringing temperatures closer to set point make occupants more comfortable?[33] The resulting decrease in temperature swings due to intermittent loads would, no doubt, enhance comfort; whether moving the average temperature closer to set point would have a similar result is questionable: set points are chosen, to some extent, to compensate for steady-state error—lower in summer, higher in winter (Int-Hout, 1989). Thus the average temperature (and corresponding energy use) in a room with PI temperature control might not be significantly different than that in one with proportional-only control. More work in this area, in particular, measured performance of installed terminal boxes with PI control, might shed light on how PI room temperature control influences comfort and energy use.

AUTOMATIC REBALANCING

Field observations described earlier showed zones in the study building in which terminal boxes were not balanced or sized properly—the maximum flow rate and/or controller gain were not sufficient to control temperatures. This situation can easily arise in office spaces, where changing tenancy might introduce loads not originally foreseen or accommodated for, or changes in partition wall configuration results in zone reorganization.[34] In the case where a box is simply too small to handle the loads in a zone and temperatures are intolerable, the only solutions are to replace the box or duct additional air from another terminal box. Although DDC control may help to identify a zone that is chronically out of control, it offers no new options here. In cases where zone temperatures are too

[32]According to two manufacturers (Int-Hout, 1989; Warren, 1989), the error is generally negative in the cooling season and positive in the heating season. Incidentally, the heating season set point is generally lower than the cooling season set point.
[33]Manufacturers seem to be in disagreement on this, depending on whether their products have PI room temperature control capability.
[34]Both of these conditions arose in the study building.

high, but the maximum flow rate is not set at the maximum available, DDC terminal boxes offer advantages.

As an example, consider a single-room zone in which, under typical peak loads, room temperature is constant at about 4 °C above set point (21 °C), and the box outlet temperature with the supply air at maximum is 15 °C. With a throttling range of 1.7 °C, we wish to reduce the temperature 2.3 °C (i.e., the room will be 1.7 °C above set point under peak load, and flow will be at a maximum). To determine how much to increase the maximum flow rate, we use Equation 3.1.1, setting the derivative to zero and rearranging:

$$T_r - T_o = \frac{\dot{Q}_l}{\rho \dot{V}_o c} \tag{3.4.1}$$

With the box outlet temperature and load held constant, we wish to decrease the left side by 2.3/10 or 23%. The desired outlet flow rate is then $\dot{V}_o / (1 - 0.23) \approx 1.3\dot{V}_o$, or an increase of 30%. Because Equation 3.1.1 is a simple representation of the heat flows in a room, this gives us only a crude estimate of the required increase in flow; nevertheless, it is a good starting point.

In a building using pneumatic terminal boxes, unless the temperature of the problematic zone is monitored by an energy management system, identifying the zone will depend on the occupant reporting the problem to building management personnel. Increasing the maximum flow rate requires doing essentially everything that is entailed in rebalancing the zone: a professional air balancing contractor must go into the ceiling above the zone, adjust the controller, and increase the speed of the motor while measuring the flow rate, so that the total flow through the fan is equal to the maximum desired flow rate. The DDC terminal box used in this study simplifies this procedure considerably—the problem zone can be identified remotely,[35] possibly before occupants complain; the maximum flow rate

[35] In multi-room zones, a problem room that does not contain a thermostat cannot be identified remotely. Once it is identified, however, the same procedures apply. Multi-room zones where the temperature in a

can be adjusted remotely as well. However, increasing the fan speed still requires getting into the ceiling, as the fan is a constant-volume (i.e., manually adjustable) type.[36]

Terminal boxes about to come on the market at the time of this writing (Warren, 1989) will include the option of a variable speed fan,[37] the speed software-adjustable, making remote rebalancing possible.[38] Such boxes will lend themselves well to duty cycling, as the fan can be cycled off and on gradually without attracting the occupant's attention.[39] Practical implementation of remote rebalancing is complicated by technical details such as automatic measurement and calibration of flow through the fan. Measuring total flow with a dynamic pressure sensor downstream of the box is rarely possible, as there is often little distance for the flow to develop. However, since flow through the fan should be nearly proportional[40] to the square root of the static pressure downstream of the fan, one way to measure this flow is to measure static pressure at this location[41] and compute a flow rate based on a constant of proportionality determined from a single pair of flow and pressure measurements made at installation. In order for this method to work, changes in the downstream ductwork that would alter the flow resistance would result in inaccurate measurements of flow. Another method is to measure secondary air volume with a dynamic pressure sensor installed in the secondary inlet. This type of sensor would not work in the long openings of the sort used in boxes incorporating noise reduction features in the

single room is out of control will require adjustments of the diffusers in each room once the flow rate is increased, so that the additional air will go to the room in need rather than over-cooling the others.

[36]Increasing maximum flow beyond without increasing the fan speed will cause supply air to spill into the ceiling plenum.

[37]Unlike the variable speed drives described in Chapter 2, the sort used here are controlled with SCRs, and do not use less energy as speed is reduced. They may, however, enable energy savings by facilitating duty cycling, as the fans may be turned off and on gradually.

[38]This particular product is designed for manual remote balancing performed by an operator.

[39]It is questionable whether comfort or ventilation constraints would permit zone-level duty cycling for demand limiting. One controls manufacturer prefers to limit demand by resetting zone temperatures (upward), thereby reducing flow and fan power.

[40]The flow resistance coefficient for duct that branches to several diffusers will not be perfectly constant with flow, but close enough.

[41]This, in itself, it not trivial—the flow downstream of the fan is not at all uniform, and a special shielded sensor that ensures dynamic pressure will be zero is required.

plenum inlet, such as that described in this study; new designs successfully incorporating this feature have symmetrical plenum inlets (Warren, 1989). Accuracy at low flow rates is not a problem in these implementations;[42] the sensor is not used for reset control, rather only to measure the secondary flow when the damper is closed.

Assuming that a practical means of measuring total flow can be implemented, control of the fan would require the addition of another control loop, perhaps similar to that controlling the damper. This would increase the burden on the microprocessor, but not beyond feasibility. Other aspects of implementing a rebalancing algorithm in real time that must be considered are:

- What should be the criteria for identifying a box in need of rebalancing? Can adjustments in gain (via throttling range) solve some poor temperature control problems?
- For the same reasons that a box's maximum flow rate might be too low, there will be those for which the opposite is true. It would be desirable to reduce the flow through these boxes for economy at the supply fan, and for comfort in other zones, if the fan is near capacity. How can "under-balanced" boxes be identified?
- Increasing the flow to a single zone will increase the total supply air flow. Should a "supply air budget" be maintained, i.e., should the sum of maximum zone flow rates be held constant, by decreasing the maximum flow to other zones? If so, how should this reduction be allocated?
- How, specifically, should the maximum flow to a zone be increased or decreased (e.g., incrementally, all at once)? How much should it be increased?
- To what extent will diverse load conditions among the rooms in a zone require changes in flow to be accompanied by mechanical adjustment of diffusers?

In particular, the problem raised by this last question may make real time rebalancing impractical in many situations. Clearly, though, more research is needed before new technology can bring this feature to the market.

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[42] Low (or slightly negative) secondary air velocities are occasionally seen when a box is providing full cooling.

Chapter 4

Supply Fan Control for Static Pressure Minimization Using DDC Zone Feedback

4.1 INTRODUCTION

In Chapter 2, we discussed the advantages to a control strategy that would allow regulating duct static pressure at the lowest possible level necessary to meet the requirements of the terminal boxes at all times, in DDC systems with VSDs. In this chapter we will examine some algorithms that might be used in control system software to implement this sort of control strategy, given the hardware capability to accomplish static pressure reset control.

Earlier, it was mentioned that experimenting with varying static pressure in the study building by manually adjusting the set point was difficult, for several reasons:

- feedback of terminal box damper position was not available at or near the location of the control panel, making it difficult to gauge the effect of changes in pressure on the box
- only one terminal box in the building was monitored, and it is likely that others were starved for air while this box was satisfied
- all of the boxes in the building except the experimental one were pneumatically controlled, and not truly pressure-independent, causing supply air flow rate to vary with changes in static pressure more than it would in a system using only DDC terminal boxes

Thus simulation is an attractive way of exploring the details of control strategies for reducing static pressure. Here we shall simulate a simple system consisting of a supply fan, a duct system, and a single zone, based on the building used in this study.

4.2 TWO FAN CONTROL APPROACHES

Two types of algorithms are proposed here: a "classical"[1] or modified PI control algorithm, and a heuristic algorithm. Both approaches regulate either static pressure or fan speed directly, using an error signal derived in some fashion from the primary flow error signal from one or more zones. A block diagram of the controller in context is shown in **Figure 4.1**. PI control was chosen as a basis for the second approach not because of any inherent suitability to this problem, but because supply fan static pressure control is traditionally and typically done using this form (Walker, 1984; Pinnella, 1985).

For each controller, it is desirable for the action of the supply fan speed control to be much slower than the terminal box controller/actuators, so as to allow the boxes time to respond to changing duct pressures and to avoid instability, yet fast enough to minimize static pressure so that energy savings may be achieved for most of the day in a system that operates diurnally. In such a system there is a cool-down period at the start of each day in the cooling season when the fan must run at full speed in order to maintain duct pressure until the terminal box dampers begin to close (as the zones cool down). Many systems such as that in the study building use an "optimal start" procedure, where interior and exterior temperatures are used to calculate when the fans and chillers need to turn on in order to cool the space by the beginning of the working day. Static pressure in the study building was typically up to set point by this time, indicating that a static pressure reduction algorithm could begin working during the cool-down period in the cooling season. During the heating sea-

[1] After adding the various computational features to compute an error signal, a deadband, and limit the output, this controller can hardly be called "classical"—the usage here is shorthand to distinguish it from the heuristic controller, and to connote the controller's PI lineage.

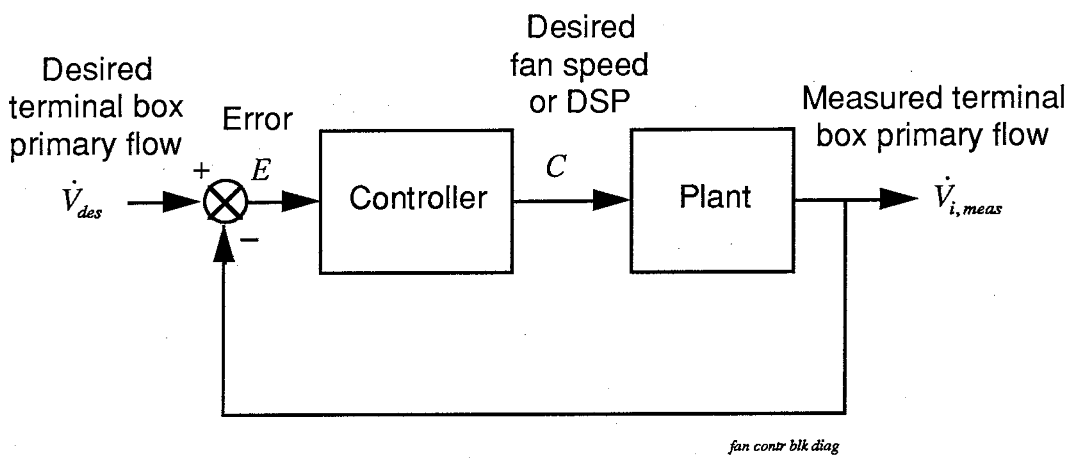

Figure 4.1. Block diagram of proposed fan controller. The plant in this case consists of the supply fan, distribution ducts, and terminal box. The error may be computed as the largest positive error for all boxes, an average of all positive error signals, or perhaps a moving average of the largest positive error.

son, there is an equivalent warm-up period during which the zones warm up. Static pressure set point is reached sooner than during cool-down, as terminal box dampers are at a minimum, and a static pressure reduction algorithm could likely begin working earlier.

There are several methods for deriving a static pressure (or speed) control error signal, computed once each polling cycle of the controller:[2]

1) The error signal $\dot{V}_{des} - \dot{V}_i$ that is largest and positive.[3] This method has the advantage of guaranteeing that all zones are satisfied. Possible disadvantages of this method are that it might result in a "bumpy" signal, leading to instability; also, a box that is grossly undersized or under-balanced will tend to dominate the system.

2) An average of all positive error signals. This method would probably result in a smoother signal than method 1, and not allow undersized boxes to dominate. The disadvantage is that the zones these boxes are in are likely to overheat.

3) A moving average of the largest positive error signal. This method would result in a smooth and more slowly changing error signal, and would foster stability. Disadvantages are that an undersized box would dominate, as in method 1, and

[2]This controller would most likely sample the zones from stored data at a rate that is asynchronous with that of the primary zone communication device or gateway that is polling the zones and storing the data.
[3]Some terminal box control programs set a flag, referred to as a "low flow alarm," when the measured flow is less than desired. Such an alarm condition could facilitate identification of such boxes and reduce the computational burden on the zone management software.

the "inertia" of this method would slow down the fan system response, depending on how many samples are averaged, and the sampling (polling) frequency.

Using a deadband to translate this error signal into an effective error, as in Equations 3.3.6, would help to stabilize the controller. Alternatively, the error for each box could be stored, as well as its time history, and the manner in which such data were used would be specific to the controller algorithm. While the issue of error signal computation will not be dealt with in depth here, the problem with any such scheme is that once all the boxes have adequate inlet pressure and the actuators have stabilized, the error signals will not be negative, and so a PI type controller using this as an input could increase static pressure, but not decrease it, and thus would not work. The control approaches described below are formulated to contend with this problem.

CLASSICAL APPROACH

The continuous-time PI controller equations 3.1.5 - 3.1.8 may be modified to include a constant (positive) decay term K_o in the integral portion of the output:

$$\frac{dC}{dt} = \frac{C^{ss} - C}{\tau}$$

$$C^{ss} = K_p E' + I$$

$$E = \dot{V}_{des} - \dot{V}_{i,meas} \qquad \left(\dot{V}_{des} - \dot{V}_{i,meas} > 0\right)$$

$$E = 0 \qquad \left(\dot{V}_{des} - \dot{V}_{i,meas} \leq 0\right)$$

$$\frac{dI}{dt} = K_i E' - K_o \tag{4.2.1}$$

where E' is an effective error computed from E using Equations 3.3.6. This controller thus uses positive error to increase static pressure, and the decay term to decrease it, bringing the system to the lowest desirable p_s. Since the gain parameters affect the rate of change of controller output, they must be small enough to allow the terminal box damper actuators to adjust to changes in p_s and desired flow rate. The discrete-time equations actually used to simulate the controller are similar to Equations 3.3.5:

$$C^{ss}_{k+1} = K_P E'_k + I_k - K_o \Delta t$$

$$I_k = \beta I_{k-1} + (1-\beta) C^{ss}_k$$

$$\beta = \frac{(2T_I - \Delta t)}{(2T_I + \Delta t)} \tag{4.2.2}$$

In addition, the output of the modified PI controller is limited between a specified minimum and maximum.

HEURISTIC APPROACH

The method presented here uses an algorithm that, on each polling cycle, tests the error and change in error for all the boxes, and decides whether to increase, decrease, or hold static pressure:

```
IF for any box E' > 0 THEN
     IF for those boxes ΔE' ≥ 0 THEN
          increase DSP set point a small amount
     ELSE
          do not adjust DSP set point
ELSE
     decrease DSP set point a small amount
```

Here E' is an effective error, as in the preceding approach, and $\Delta E'$ is the effective change in the effective error, determined using a deadband as in Equations 3.3.6:

$$\Delta E_{k+1} = E'_{k+1} - E'_k$$

$$\begin{array}{ll} -d \le \Delta E_{k+1} \le +d: & \Delta E'_{k+1} = 0 \\ \Delta E_{k+1} > d: & \Delta E'_{k+1} = \Delta E_{k+1} - d \\ \Delta E_{k+1} < -d: & \Delta E'_{k+1} = \Delta E_{k+1} + d \end{array} \tag{4.2.3}$$

where k indicates the value at the previous polling cycle. The rates at which the set point is increased or decreased are not necessarily equal. We shall denote the increment rate as K_i

and the decrement rate as K_d. The increase or decrease in set point is then $K_i\Delta t$ or $K_d\Delta t$, where Δt is the sampling period.

4.3 THE SYSTEM MODEL

The terminal box/zone model was augmented to include a supply fan and air distribution system. Again, the intent here was not to produce a model that accurately represented the short time scale (i.e., seconds) thermal and fluid dynamics of the fan and ducts, but one that would serve reasonably well for preliminary evaluation of the proposed control algorithms. Thus various aspects of the supply system are neglected: the cooling coil and mixing valve, the rotational and electrical dynamics of the fan and motor, and time delays in propagation of duct pressure due to duct length. These features have been modeled in other studies of VAV systems, notably Percivall (1985) and Pinnella (1985). In the latter, however, the author concludes that of these, only duct time delay and tubing dynamics (in pneumatic systems) were important, and then only in systems where ducts and tubing are long. The time scale of interest here is minutes, not seconds, and although the duct lengths are long, this factor is not likely to be significant.

Several approaches to representing the supply air system using *HVACSIM+* were tried. The models differ in the way in which they represent unmodeled zones, but all use the same basic supply fan and control system. This consists of a fan component from the basic library, the same fixed-rate actuator by Haves (1985) to operate the damper, and a motor speed control and control element, both developed in the course of this research. The configurations used here control fan speed directly rather that static pressure. **Figure 4.2** shows how the control system components are connected.

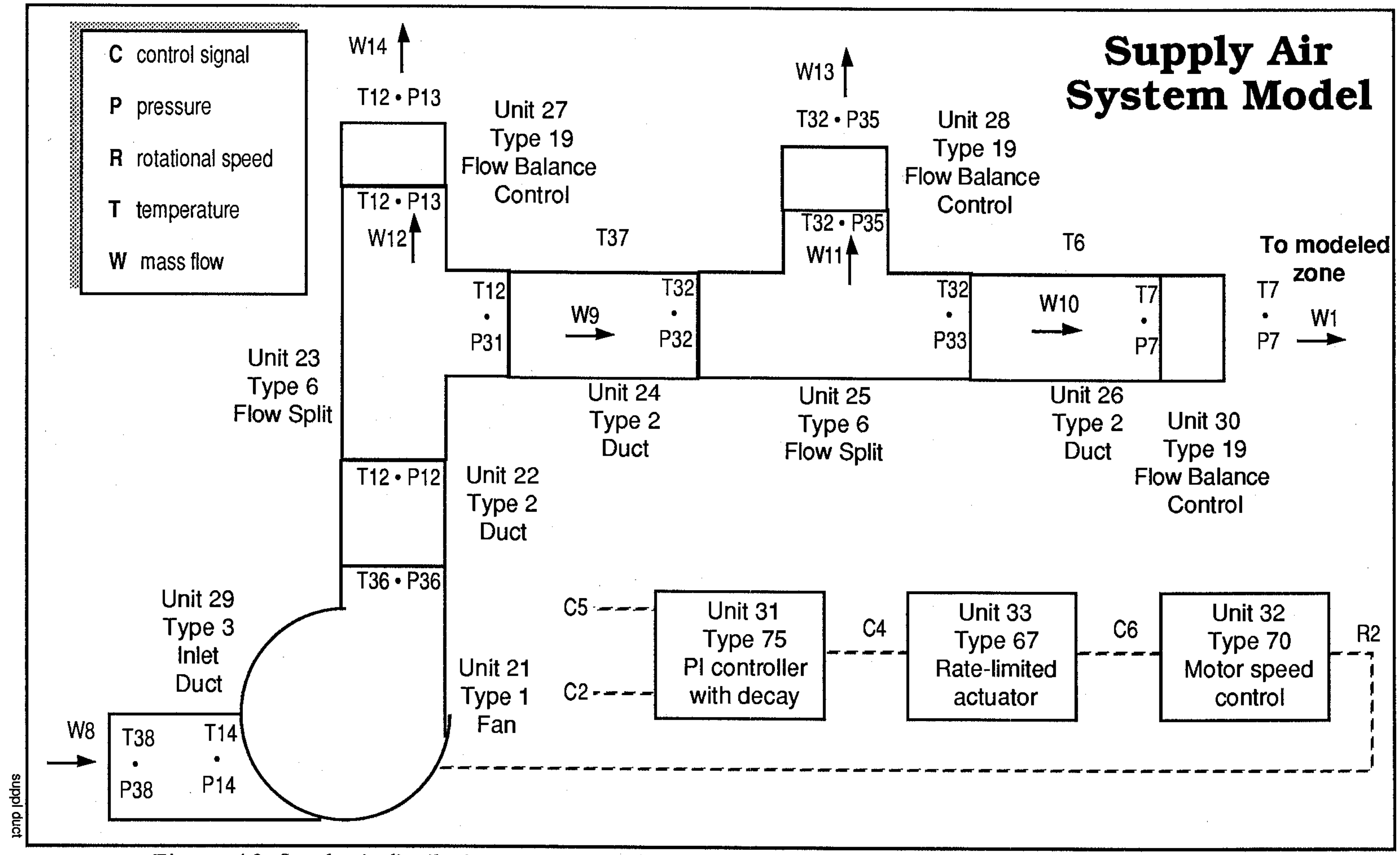

Figure 4.2. *Supply air distribution system model, including fan controller, for use with HVACSIM+. In this model, flow to unmodeled zones is held constant by the flow balance control components. These components have no physical analog; they are used in the simulation to enable specification of flow, rather than pressure, as a downstream boundary condition.*

In these simulations fan speed was controlled directly, rather than controlling static pressure with an inner fan speed control loop. This was done both for simplicity and to reduce the computation requirement. The control element, an implementation of either Equations 4.2.2 or the heuristic algorithm described above, computes an error signal and an output that serves as input to the fixed-rate actuator, which represents the ramping control in a variable speed drive. The travel time for the actuator was set to 300 s, the maximum ramp time for the VSDs in the study building. The motor speed control component simply converts the actuator output control signal to a rotational speed, using a single gain parameter. The source code for the controllers and speed control is given in Listings C.4 through C.6 in Appendix C.

In an attempt to make the distribution system model represent an actual system, it was based on the study building. *HVACSIM+* determines the fan inlet pressure (in kPa) from the outlet pressure as follows (Clark, 1985):

$$p_i = p_o - 0.001 C_h \rho N^2 D^2 \tag{4.3.1}$$

where

N	=	rotational speed (rev/s)
D	=	impeller diameter (m)

This equation defines the dimensionless pressure (head) coefficient C_h. This coefficient is determined as a polynomial function of the dimensionless flow coefficient,

$$C_h = a_0 + a_1 C_f + a_2 C_f + a_2 C_f + a_3 C_f \tag{4.3.2}$$

which is defined as

$$C_f = \frac{\dot{m}}{\rho N D^3} \tag{4.3.3}$$

where $\dot{m}$ is the mass flow rate in kg/s. The coefficients of the pressure/flow curve are input as parameters, and were determined here by fitting a third degree curve to manufacturers' data as

$$a_0 = 1.356$$

$$a_1 = 0.255$$

$$a_2 = -2.27$$

$$a_3 = -40.01$$

The fan outlet temperature is determined from the inlet temperature (held constant for these simulations) using conservation of energy:

$$T_o = T_i + \frac{(p_o - p_i)}{\rho c}\left(\frac{1}{\eta} - 1\right) \tag{4.3.4}$$

where the static efficiency η is determined a polynomial function similar to the pressure coefficient:

$$\eta = e_0 + e_1 C_f + e_2 C_f + e_2 C_f + e_3 C_f \tag{4.3.5}$$

Since manufacturers' efficiency curves for this fan were not available, a quadratic efficiency curve was fit to values derived from values of flow, pressure, and power from the fan curves using the definition of static efficiency (ASHRAE/AMCA, 1975):[4]

$$\eta = \frac{\dot{V}\Delta p}{P_f} \tag{4.3.6}$$

where the pressure rise Δp is in Pa, the flow $\dot{V}$ is in m^3/s, and the shaft input power P_f is in W. The coefficients are then

$$e_0 = 0.0 \text{ (constrained)}$$

$$e_1 = 10.15$$

$$e_2 = -31.98$$

[4]The compressibility factor is omitted, as it may be taken as unity when the total fan pressure is less than about 3 kPa (12 in. H_2O).

The first branch in the duct represents the first floor take-off; the second branch represents the take-off serving the modeled zone. Parameters were based on information taken from drawings and available measurements, and adjusted during successive simulation until a reasonable representation was achieved.

In an actual air distribution system, the steady state response of total system flow to changes in fan speed is not simple. As discussed in Chapter 2, a system in which zone flow rate is not controlled to set point (and is therefore not proportional to the difference between zone temperature and set point) will display a decrease in total supply air flow rate as duct static pressure (or fan speed) is decreased. This will not occur in a system using terminal boxes such as the one described here—where zone flow rates are proportional to temperature error signals—as long as none of the terminal box dampers are completely open or closed. In an actual system, however, where zone capacity/load ratios are diverse, it is likely that at any given time some dampers would be at the limit of their range, causing total flow to decrease somewhat as static pressure is reduced. At both ends of the static pressure operating range, this effect would be more marked. Such a relationship between flow and static pressure is shown in **Figure 4.3**. The exact nature of the response for a particular system would vary with load on the zones. Thus an accurate model for testing supply fan control systems would necessarily include many zone models with a range of characteristics. It is conceivable, however, that a single component model (a damper of sorts) could be devised to represent the response characteristic of a whole system.

Implementing either of these alternatives was beyond the scope of this study. There were two potentially feasible ways in which to represent the effect of unmodeled zones on the distribution system: as one or more constant flow resistances, or one or more constant flow rates. The flow through the modeled zone is a small fraction of the total, varying between about 1% and 5% of the average total flow. Using a constant resistance for the unmodeled zones causes duct static pressure to be very sensitive to the modeled zone damper position (for a given supply fan speed), and the zone tends to dominate the system.

A constant flow rate can be modeled using a component called a flow balance control. This component does not represent any physical part of an actual system; rather it is intended for connecting conduits between superblocks, which may not be coupled tightly enough to permit self-consistent solutions of the sequentially solved sets of flow and pressure equations (Clark, 1985). The flow balance control determines from the outlet flow rate the pressure at the connection of the components it is coupling by varying an internal flow resistance. In this case it enables specification of a flow, rather than a pressure, as a downstream boundary condition. **Figure 4.2** shows a supply air system model that incorporates these components. Using a constant flow rate for the unmodeled zones, however, causes static pressure to be largely determined by supply fan speed. Thus with no change in total flow, static pressure drops off sharply as fan speed is decreased. In an actual system, such decreases in static pressure would likely be accompanied by decreases in flow as dampers go wide open; this is not consistent with the assumption of this model that the modeled zone is the "worst-case" zone at all times. Despite its limitations, this model proved to be adequate for preliminary evaluation of the proposed control methods.

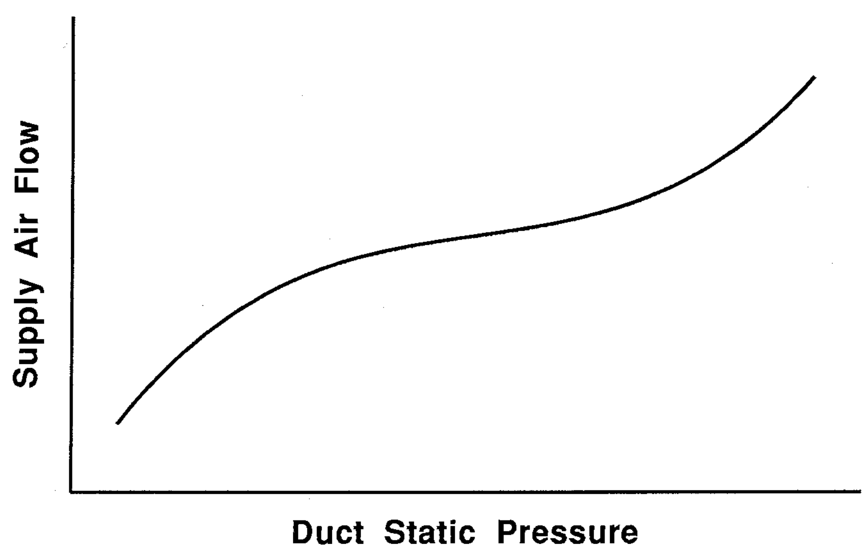

Figure 4.3. Approximation of how supply air flow rate might vary with static pressure in a VAV system in which terminal boxes control zone flow rate to set point.

One way to more closely represent the worst-case box at all times is to have the supply fan serve only the modeled zone. This can be done by down-sizing the fan model—simply reducing the diameter until the flow rate for a given pressure difference matches the requirement for the terminal box model. The fan and supply duct can be in a separate superblock as before, using a flow balance control to connect it to the terminal box, or it can be part of the same superblock, in which case the primary inlet duct on the terminal box is replaced with a regular duct model.

4.4 SIMULATION AND PARAMETER SELECTION

Simulation using a constant flow for unmodeled zones, despite the limitations described above, produced results that are useful in comparing the controllers, showing how they interact with the terminal box controls, and showing the effect of control parameter choice on system response.[5] The results of several simulations of each controller are presented here to convey a qualitative understanding of the effect of parameter changing the gain, deadband, and sampling interval parameters.

The model had to be modified slightly in order to avoid instability in the air stream temperature: the temperature of the air leaving the supply system superblock was decoupled from the terminal box inlet, making this inlet temperature a boundary variable, which was set at a constant (typical) value of 13.3 °C.[6] Fan speed was limited to a range bounded by 15.0 rev/s at the upper end (the actual limit of the fan in the study building is 14.6 rev/s), and either 10.0 rev/s or 11.25 rev/s (for the first four simulations) at the lower end. The lower bound was somewhat arbitrary, but was used in order to avoid unrealistically low static pressures caused by the constant flow constraint. As in the simulations of the zone

[5]Unfortunately, the simulations using the configuration in which a small fan serves only the modeled zone were not successful, due to numerical difficulties. The reasons for this are not clear; this and other simulation obstacles are discussed in Chapter 5. For reference and comparison, the component configuration for this model is given in Listing C.10 of Appendix C.
[6]The reasons for this instability are not known; this is discussed in further in Chapter 5.

alone, a minimum simulation time step of 0.001 s, a maximum time step of 1 s, and a reporting time of 1 minute were used.

HEURISTIC CONTROLLER

We shall look first at the heuristic controller. **Table 4.1** lists the parameter values and summarizes performance for the example simulations presented here. The choice of parameters is discussed below; the complete configuration of components is given in listing C.8 in Appendix C. Simulation results for the first simulation of this controller are displayed in **Figure 4.4**; results for the remainder are shown in **Figure 4.6** at the end of this chapter.

$K_i > K_d$. We begin, in Simulation **h1**, with low values of increment rate K_i and decrement rate K_d. The initial value of $K_d = 0.00015$ was selected in order to give a rate of fan speed reduction of (15 rev/s)(0.00015/s) = 0.00225 rev/s^2, sufficient to decrease the speed from 15 rev/s to 11.25 rev/s in about 30 min. The initial choice of $K_i = 0.0002$ is that value required for the time it takes the fan speed to change 4 rev/s to be one order of magnitude slower than the time it takes the damper to open. The controller successfully reduces static pressure and causes the box damper to open. Its action does not appear to be too fast for the terminal box, which is able to keep the flow rate within a few percent of desired, except following step changes in heat gain, when the box takes from 3 to 10 minutes to catch up, depending on the damper position and inlet static pressure. The speed-decreasing action of the controller is, however, undesirably slow; it takes several hours to open the damper to near capacity. At this rate, it would take 9.5 hours to go from fully open to closed continuously. One might hastily conclude that K_d is too small; this rate is much slower, however, than the rate at which the fan speed is actually reduced by the controller, 0.00225 rev/s^2 (the steepest negative slope of the fan speed line in the first figure for this simulation). Likewise, the effective rate of damper opening is much slower that the fixed rate of the damper actuator, which takes 180 s to travel limit to limit.

Oscillatory response. The hunting seen in fan speed (and its effect on inlet pressure) is one of the keys to what is going on. Although the time scale of these graphs is too large to examine controller interaction in detail, it can be seen that the damper does not display a similar pattern, ruling out controller interaction. Another look at the algorithm explains these oscillations: fan speed and static pressure decrease, causing flow to decrease, resulting in a positive error E' and positive $\Delta E'$. The controller sees this and responds by increasing fan speed, causing flow to increase. Although E' is still positive, $\Delta E'$ is now negative, and fan speed is unchanged. Eventually, the flow increases until the error is zero or less, beginning the cycle over again. The oscillations during the period from hour 13 to 14 occur about 17 times per hour, giving each cycle about 8 sampling intervals of the controller. The oscillations at other times are not as regular.

Table 4.1. HEURISTIC FAN CONTROLLER PARAMETER VALUES AND PERFORMANCE

Simulation	Inc. rate K_i (s^{-1})	Dec. rate K_d (s^{-1})	Deadband d	Sampling Period Δt (s)	Flow[a] at Zone	Pressure[b] Reduction
h1	0.00020	0.00015	0.008	30	☺☺	☹
h2	0.00030	0.00030	0.008	30	☺☺	☹
h3	0.00030	0.00045	0.008	30	☺☺	☹
h4	0.00045	0.00045	0.008	30	☺☺	☹
h5	0.00020	0.00015	0.008	60	☺☺	☹
h6	0.00020	0.00015	0.020	60	☺	☺
h7*	0.00020	0.00015	0.020	90	☺	☺☺
h8	1.5E-5	0.00015	0.008	30	☹	☺☺
h9	7.5E-6	0.00015	0.008	30	☹	☺☺

*Best performer

[a]Ability to provide desired primary air flow at zone

[b]Ability to reduce duct static pressure

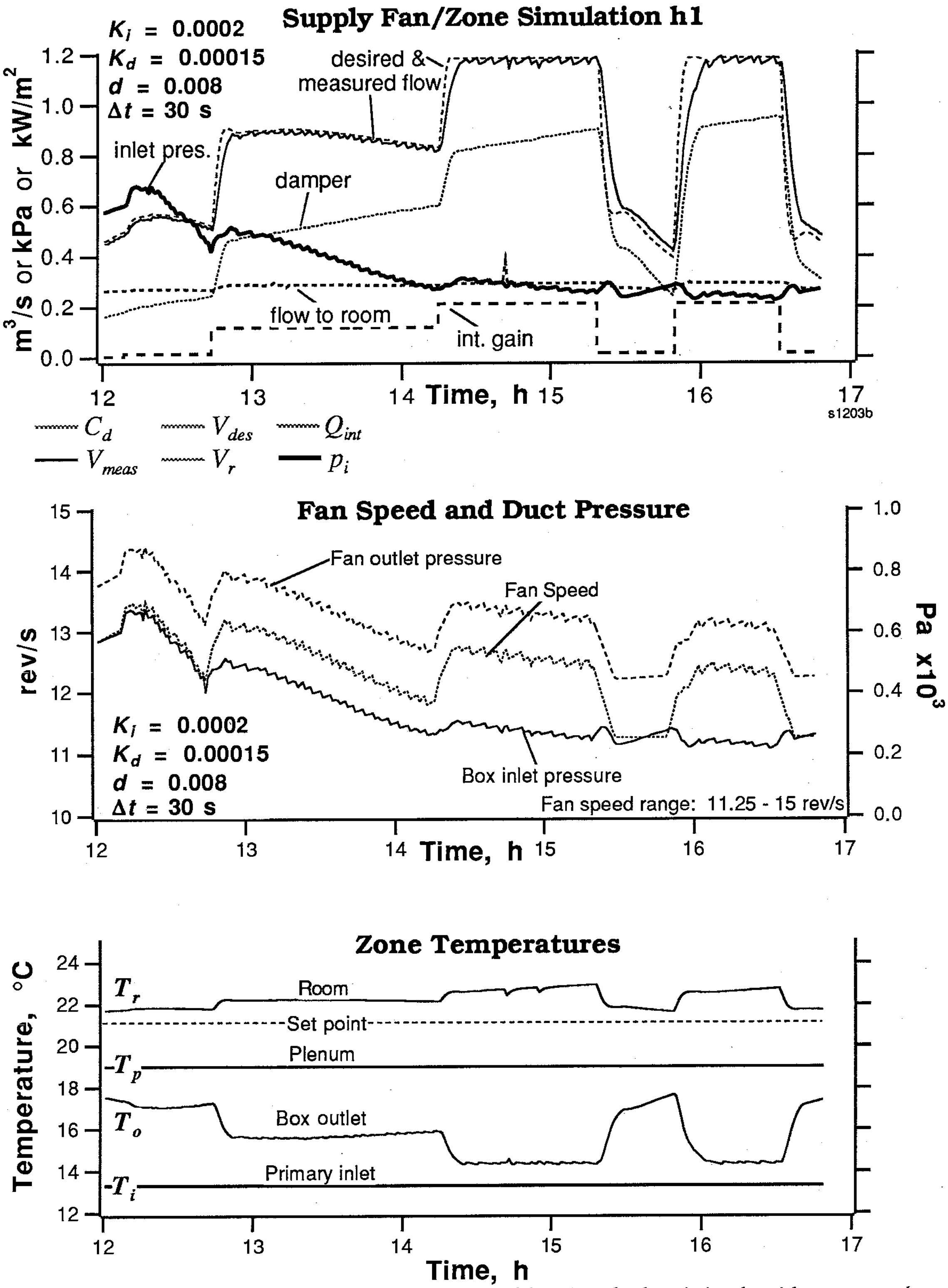

Figure 4.4. Results of supply fan and zone simulation h1, using the heuristic algorithm proposed in this study. Results of the remaining runs for this controller are included in ***Figure 4.6*** *at the end of the chapter. Increment and decrement rates, deadband, and sampling period are varied across simulations. Internal gain (not shown) is the same as in the previous simulations of the zone. External gain is normalized by floor area of the room receiving the gain. Temperatures are similar for all the simulations, and so are shown only for the first.*

Another notable feature of Simulation h1 is the period between hour 15 and 16 when fan speed reaches and maintains its minimum value for about half an hour, while the load is off. As desired flow rate decreases with room temperature, the damper closes and inlet pressure rises. This is an artifact of the artificially constant flow rate through the rest of the system; if this box were indeed the "worst case" box in a real system, there would be no reason to keep static pressure from dropping further at this point.

A plausible way to reduce the hunting problem is to decrease K_i with respect to K_d, either by decreasing K_i directly or by increasing K_d. As $K_d = 0.00015$ should be sufficient to reduce static pressure rapidly, reducing K_i is a more promising choice. First, though, it is useful to look at the results of increasing both K_i and K_d. Simulations **h2 through h4** show the effect of increasing K_i and K_d. There are some noteworthy differences between cases in which $K_i = K_d$ (h2 and h4) and those in which $K_i < K_d$ (h1 and h3):

$K_i = K_d$: no oscillations, but limited damper range. When the increment and decrement are equal (h1 and h4), the static pressure decreases to $p_i \approx 0.3$ kPa (1.2 in. H_2O) until the damper is about 80% open, but no more; apparently the equal parameters cancel each other.

$K_i < K_d$: oscillatory, damper opens fully. In the simulations where they are not equal (h1 and h3), static pressure drops low enough for the damper to open fully (pi ≈ 0.2 kPa or 0.8 in. H_2O); both also exhibit the regular oscillation described above, absent in Simulations h2 and h4.

For the remaining simulations, the minimum fan speed was set to 10.0 rev/s, as this lower minimum appeared to be satisfactory in Simulations h2 and h4. The parameters of Simulation **h5** are identical to those of Simulation h1, except for the controller sampling period Δt, increased to 60 s. Although static pressure decrease and damper opening occurs slightly faster than in Simulation h1, the results are similar. As can be expected with a longer sampling interval, the amplitude of the oscillations in fan speed and flow has increased; the average steady state error in flow is still small. The more rapid pressure decrease can probably be attributed to the greater oscillation, which causes the measured flow to exceed desired more often. As in Simulation h4, there is a spike in inlet pressure (and a corresponding drop in damper position) just after hour 15, most likely a difficulty in convergence encountered at this point in the execution of the program. This also occurs at the end of Simulation h7.

Increase deadband: pressure decreases faster. In Simulation **h6** we increase the deadband d from 0.008 to 0.02 while leaving the sampling period at 60 s.[7] This greatly increases the speed at which static pressure can be reduced and the damper opened—less than two hours into the simulation, the damper is fully open. It is evident that the fan controller is allowing the damper to adjust to changes in inlet pressure; the fan controller plays much less of a direct role in controlling the zone flow rate. The controller is slow to respond to step increases in desired flow rate; the response takes from 5 to 20 minutes, depending on inlet pressure and damper position. Close examination of periods during which this occurs shows that during alternate sampling intervals, the fan controller does not increase speed at all, seeing a decrease in flow rate error due to its action during the previous sampling period. Despite inadequate flow to the zones during these periods, room temperature (not shown) is not noticeably different from that for Simulations h1 or h5. The behavior of the controller does suggest possible improvement in the algorithm to enable fan speed to increase consistently when the flow rate error is positive, although this would have to be implemented in such a way so as not to usurp primary control of the flow rate from the damper controller.

Increase sampling period: pressure decreases faster, oscillation increases. Increasing the sampling period further to 90 s in Simulation **h7** increases the rate at which static pressure drops; the damper is open within an hour. The price for this, however, is increased amplitude of oscillation.

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[7] Increasing the deadband alone was tried, however the simulation failed because of numerical difficulties.

Small K_i: pressure decrease fast, pressure increase too slow. In simulations **h8** and **h9**, small values of K_i were used (all other parameters are the same as in Simulation h1). The controller is able to reduce static pressure rapidly; the damper is fully open in one hour for h9 and two hours for h8. While the damper is not yet fully open, the steady state error is small; however when the damper is open, the fan controller is slow to respond to step changes in desired flow rate. Room temperature overshoot is more marked in these simulations than the previous ones.

MODIFIED PI CONTROLLER

Parameter values for simulations performed with the modified PI fan controller model are summarized in **Table 4.2.** Component configuration is given in Listing C.9 in Appendix C. Simulation results for the first simulation are displayed in **Figure 4.5**; results for the remainder appear in **Figure 4.6** at the end of the chapter. The simulations show results for several combinations of gain, deadband and sampling period. For each of the first three simulations, the integral time K_P/K_I is the same—about 15 s—the lowest it could be given a sampling interval of 30 s. This is a limitation of the integration algorithm used, which is not stable for sampling periods of more than twice the integral time.[8] The effective integral gain $K_I^* = \Delta t K_I / K_P$ ("repeats per sampling interval") was 2.0 for all simulations. The output decay constant K_o was held at 0.00015.

Increasing gains reduces error, pressure reduction rate. Starting with Simulations **c1**, **c2**, and **c3**, we see that increasing both proportional and integral gains brings the flow rate error into an acceptable range. In these simulations, static pressure is reduced and the damper opens wide very quickly, relative to most of the trials of the heuristic controller. The controller of Simulation c3 performs quite well; the static pressure is decreased to the point where the damper is fully open in about one hour, there are no

[8] It would have been possible to create a model without this limitation, perhaps using backward-referenced integration; as this was not done here, controller gains were chosen to conform to this limitation.

wild oscillations in flow, and the controller responds rapidly to step changes in desired flow rate, even when the damper is fully open. Doubling the gains once more (Simulation **c4**) reduces the error further, but also doubles the time it takes to open the damper. It is clear that, in selecting gain values (holding other parameters constant), there is a tradeoff between steady state error and static pressure reduction rate.

In Simulation **c5** of this controller, we double the sampling time to 60 s, which requires doubling the integral time—the increase in integral time is most likely the cause of the resulting increase in steady state flow error, although some of the error can be attributed to the deadband, which has been increased to 0.02. The rate at which static pressure is reduced is about equal to that of Simulation c3, suggesting that one of the factors determining this rate is the ratio of K_I to K_o, the same in both of these cases.

Table 4.2. MODIFIED PI FAN CONTROLLER PARAMETER VALUES AND PERFORMANCE

Simulation	Prop. gain K_p	Int. gain K_I (s^{-1})	Decay K_o (s^{-1})	Dead-band d	Samp. Period Δt (s)	K_I^*	Flow at Zone[a]	Pres-sure Red.[b]
c1	0.0376	0.0025	0.00015	0.008	30	1.99	☺☺	☹☹
c2	0.0751	0.0050	0.00015	0.008	30	2.00	☺☺	☹☹
c3*	0.1510	0.0100	0.00015	0.008	30	1.99	☺☺	☺
c4	0.3010	0.0200	0.00015	0.008	30	1.99	☺	☺☺
c5	0.3010	0.0100	0.00015	0.020	60	1.99	☺☺	☺
c6	0.3010	0.0400	0.00015	0.008	15	1.99	☹	☺☺
c7	0.3010	0.0400	0.00015	0.020	15	1.99	☺	☺☺
c8	0.4510	0.0100	0.00015	0.020	90	2.00	☺☺	☺
c9	0.9010	0.0200	0.00015	0.020	90	2.00	☹	☺

*Best performer
[a]Ability to provide desired primary air flow at zone
[b]Ability to reduce duct static pressure

Increasing deadband decreases error, increases pressure reduction rate. Decreasing the sampling interval to 15 s enables decreasing the integral time to 7.5 s by increasing K_I to 0.04 (Simulation **c6**). Simulations 6 and **7** differ only in the choice of deadband. A comparison of the results makes clear the dominant effect of this parameter: approximately doubling the deadband halves the steady state error (of little consequence, as it is already small), and also doubles the rate at which static pressure is reduced. Thus it appears that for a given K_o, the rate at which static pressure is reduced can be increased either by increasing deadband or decreasing both K_P and K_I. Increasing the deadband is preferable, as it also reduces steady state error, whereas decreasing both gains has the opposite effect.

Increasing K_P and sampling rate: oscillatory. Two more simulations were performed in order to see the effects of larger values of K_P and sampling rate. In both of these cases (**c8** and **c9**), the integral time, 45 s, is longer than, and the amplitude of oscillations is greater than in previous simulations of this controller. In Simulation c9, the controller is unsuccessful at keeping static pressure at a low level.

WINNERS AND LOSERS: COMPARING THE CONTROLLERS

Tables 4.1 and **4.2** rate each combination of parameters for their ability to reduce static pressure (and open the damper) and provide adequate flow at the zone. Giving priority to the static pressure reduction criterion, configurations h7 and c3 are the clear winners. Both achieve full damper opening within an hour, and result in relatively small steady state error in flow rate. If the simulated performance is representative of a real system, either of these controllers could be used "as is" to reduce supply fan energy use. If oscillation in fan speed or zone flow rate is a concern, further tuning of the h7 parameters is necessary to eliminate this behavior. In general, the modified PI controller does not exhibit such hunting, except where the integral time is long ($K_p/K_i > 30$ s) and the sampling period is long ($\Delta t > 60$ s).

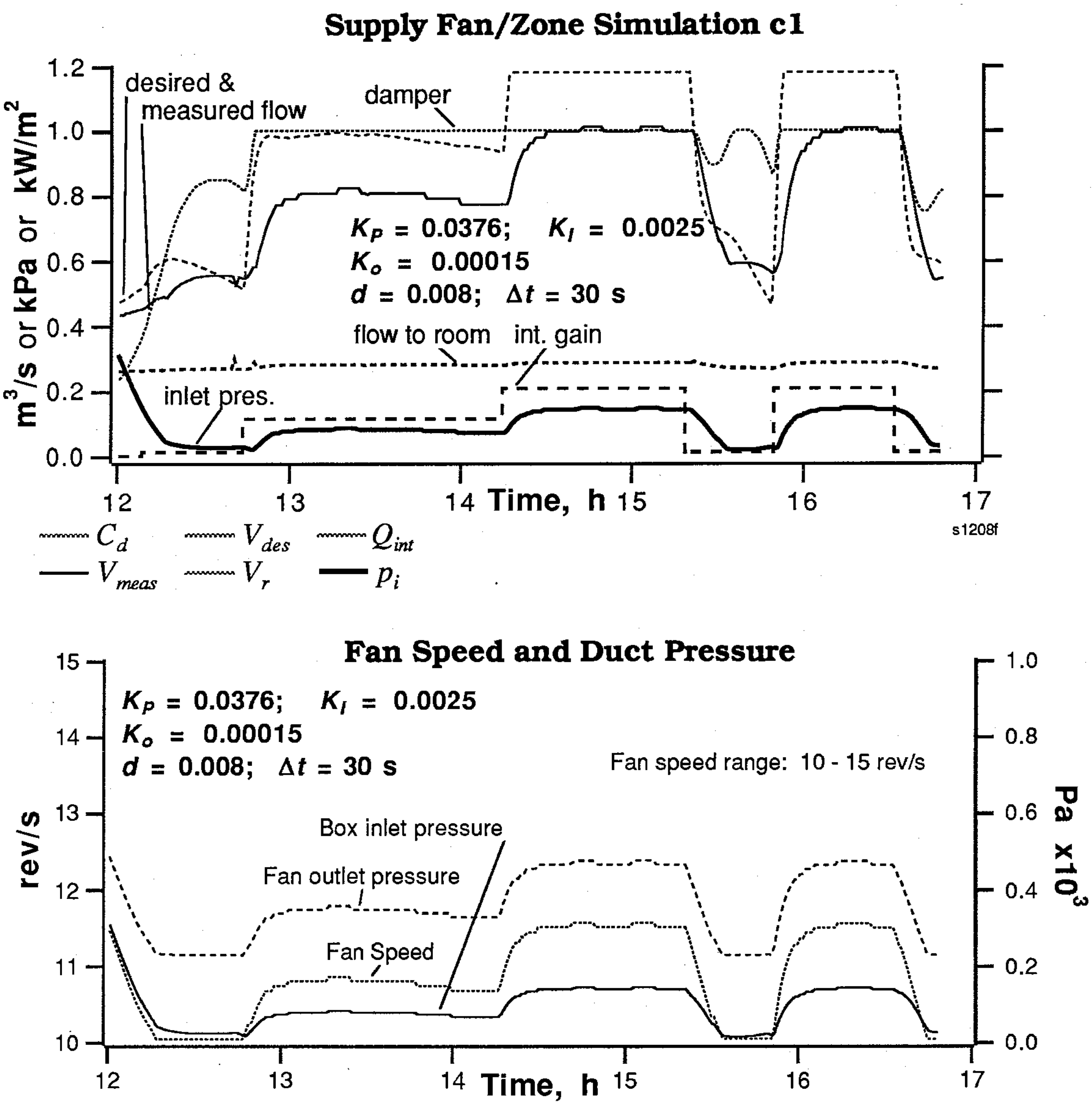


*Figure 4.5. Simulation c1, using the modified PI controller. The remaining eight runs for this controller (c2 through c9) are included in **Figure 4.6** at the end of the chapter.*

Simulation h6 of the heuristic controller and c4 of the modified PI controller are similar in both steady state error and time required to open the damper, and therefore serve as a good basis for qualitative comparison. Although the amplitude of hunting is considerably more for the heuristic controller, this is primarily due to the long sampling interval of 60 s, evident in a comparison with other simulations of this controller. The predominant difference is the rate at which fan speed is increased: given a step increase in desired flow rate, the PI controller's response is much faster (making it more desirable); the rate of decrease is about the same. The heuristic controller gives the damper controller more time to adjust to changing inlet pressures, seen in the response to the first large step increase in load. The trade-off (for the heuristic controller) is one of static pressure reduction for deficits in flow following sharp increases in load.[9] It is plausible that the heuristic algorithm could be modified to eliminate the alternate "wait actions" under some circumstances as stated earlier, thereby reducing the response time by half. A hybrid of the two might even developed, using PI control when the error signal is greater than zero, and fixed decrement otherwise.

4.5 Conclusion: Static Pressure Minimization Control

We began this chapter by proposing two methods of VAV supply fan control aimed at minimizing duct static pressure, and therefore fan energy use. Both methods use feedback from the zone flow controllers, but take quite different approaches. The heuristic method is a computerized implementation of a simple three-step procedure that might be followed by an ideal building operator whose only purpose is to minimize static pressure while keeping zone flow requirements satisfied. The other is a discrete-time implementation of classical PI control, with a rather unconventional modification intended to continually "force" the controller output (and hence fan speed) to decrease.

While it is easy to envision the successful operation of the heuristic method, the same cannot be said for the modified PI method, as it is difficult to predict the interaction between the integral gain and decay terms; simulation is clearly indicated in this case. As it turns out, the nuances of parameter selection make simulation an invaluable tool in both

[9]The argument that such step increases in load are not typical in an office space (and even if they are, the consequences of a temporary flow deficit are not critical) can be made, nevertheless.

cases. Steady state error and pressure reduction rate prove to be quite sensitive not only to gain and decrement rate, but also to deadband.

Eighteen five-hour simulations demonstrate the effect of parameter changes, resulting in a single "best" combination for each controller, and enable qualitative comparison between the two types. Both controllers successfully minimize fan speed and (indirectly) static pressure, enabling operation of the terminal box at its maximum damper opening, while maintaining the ability to meet flow requirements that change in response to loads; the modified PI controller emerges, however, as the winner, for its more rapid decrease of static pressure, and its smoother output.

Further work aimed at practical implementation of the controllers developed here would benefit from additional parametric analysis in order to improve the understanding of the effect on performance of, in particular, decrement rate and effective integral gain. Tests done on a shorter time scale (perhaps a few hundred seconds in length) could improve tuning and enable close examination of the interaction observed here between the fan and damper controllers.

The question of the best method to sample and compute error signals must be addressed. The supply air distribution system used in the simulations is not representative of a real system in the way total supply air flow and system pressures are modeled; the simulation difficulties encountered here must be solved in order to overcome these limitations of the model. Also, we have only modeled a fan powered box; many VAV systems using constant volume fan powered boxes (such as that in the study building) include variable volume boxes in less critical areas such as corridors. Boxes without fans typically have higher minimum inlet pressure requirements, and might be the first boxes to starve when static pressure is decreased. Modeling them in a simulation would lend valuable insight as to their impact on supply fan control for static pressure reduction.

Here we have used an error signal obtained from a local zone control loop to directly control the speed of the supply fan, for the sake of simplicity. In an actual implementation of supply fan control, using an inner loop to regulate static pressure and resetting the static pressure set point in the outer loop might be more desirable from a stability perspective (Warren, 1989).

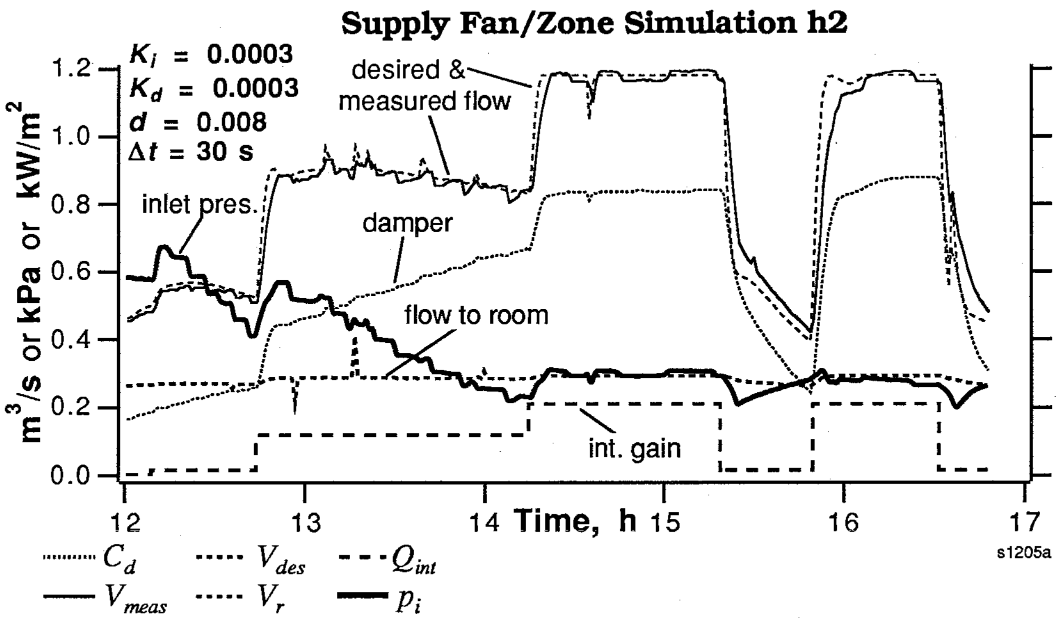
Two larger questions concern the fundamental nature of the control method used: suitability and optimality. The choice of the two algorithms tested here was somewhat arbitrary. Often PI control can be improved upon using frequency-domain analysis, with z-transforms of a linearization if the system is non-linear; in addition, such analysis can yield insight into the behavior of the system (Rohrer and Stoecker, 1986). Linearization may be unworkable for this system however, in which non-linearities like deadband and the interest in only positive error signals are so consequential.

The cost or penalty function here is supply fan energy use.[10] We have used duct static pressure as a surrogate, since energy increases with static pressure for a given supply flow rate. The relationship is not linear, however, as fan and motor efficiencies change with pressure. A controller that truly minimizes fan energy might require the use of an optimization method with energy as its cost function.

Finally, experimentation in an actual building is perhaps the only way to confront the hurdles that lie between the theoretical development of the supply fan control algorithms proposed here and their practical implementation. At least one other researcher has initiated such a program at the time of this writing (Norford, 1989).

[10]A building manager might consider the cost to be the actual cost of electricity, often complicated by demand and consumption billing structures set by the utility.

ADDITIONAL FIGURES



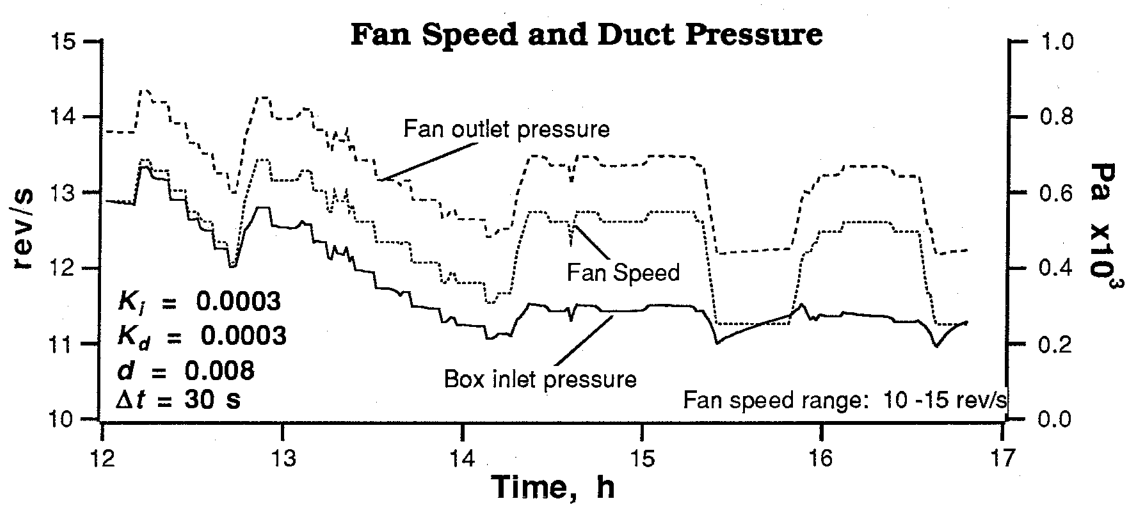
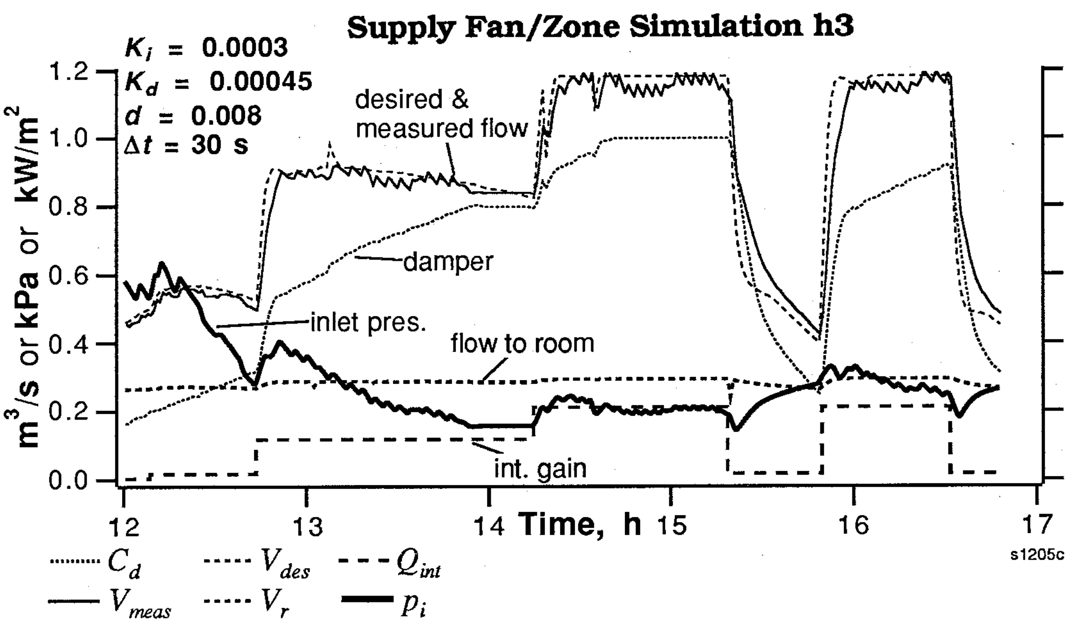
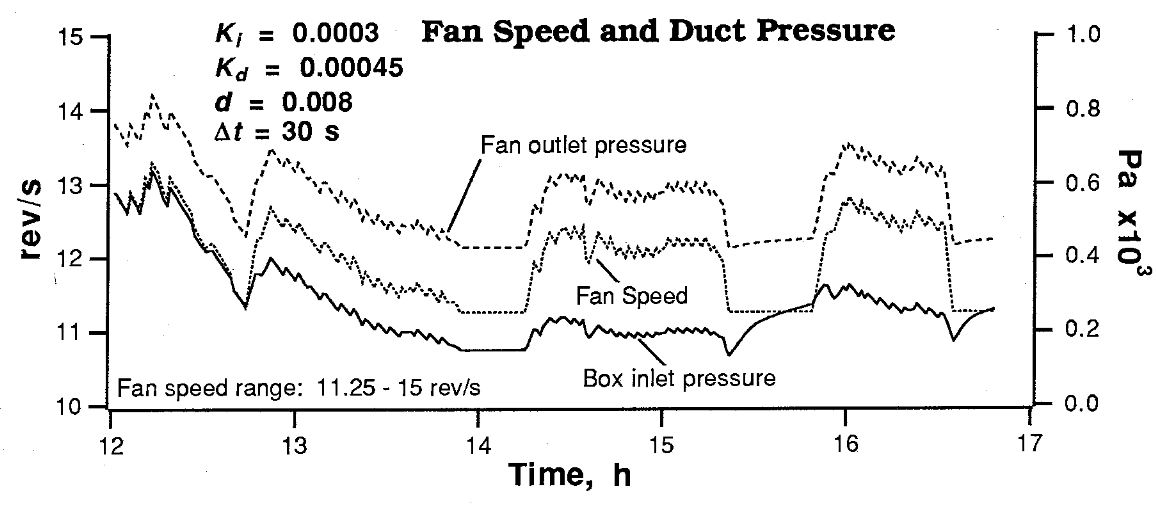


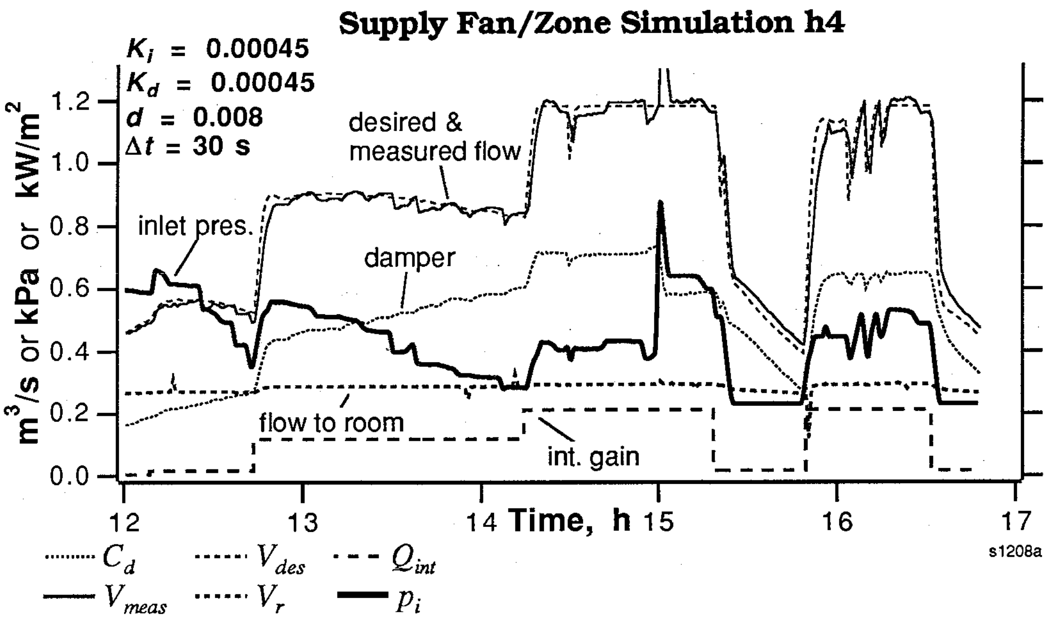
Figure 4.6. Results of simulations h2 through h9, followed by c2 through c9.

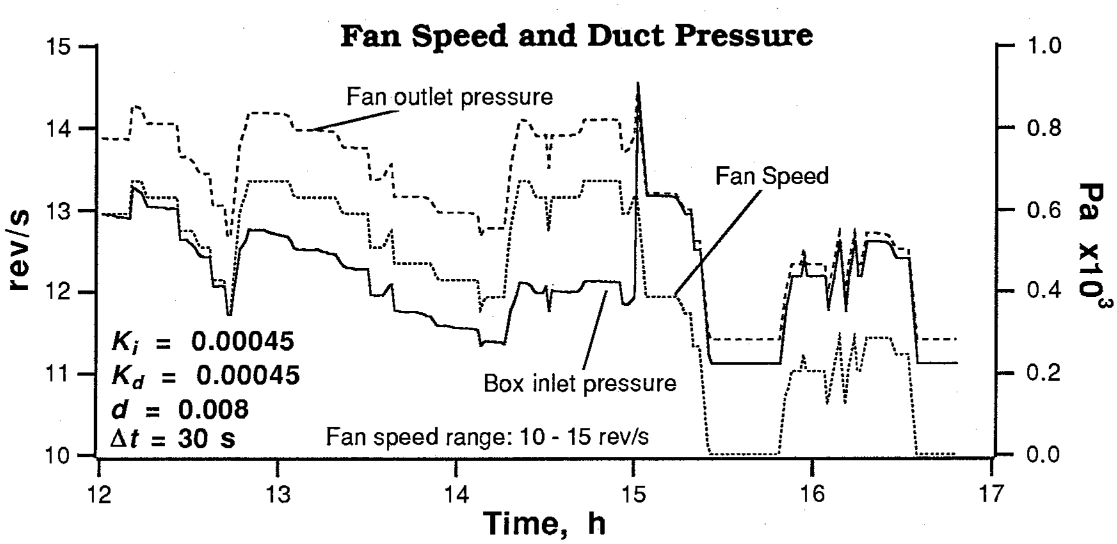


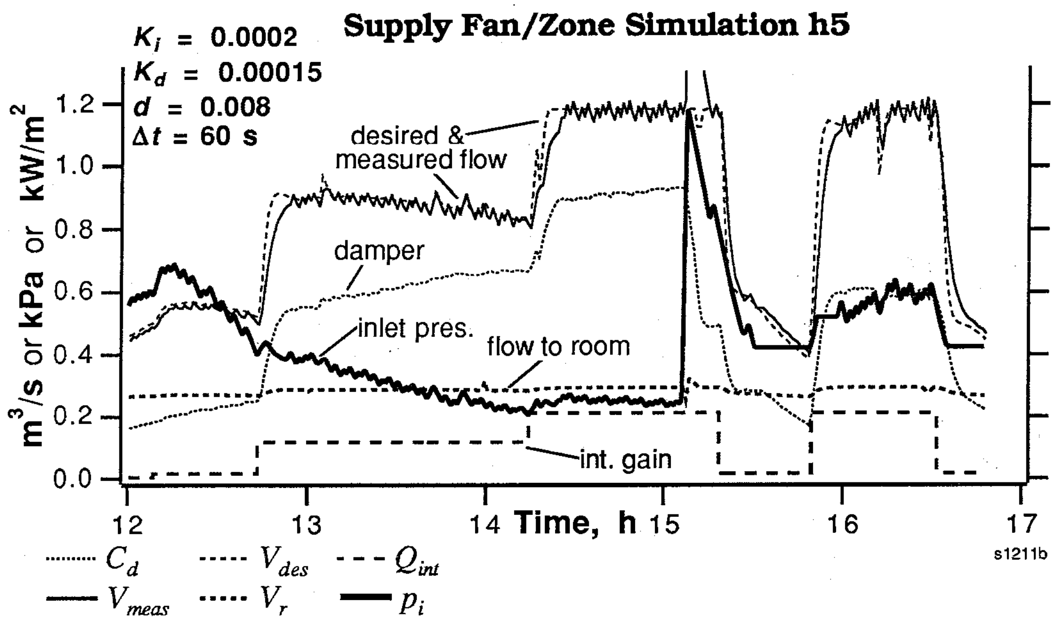
Supply Fan/Zone Simulation h3
K_i = 0.0003
K_d = 0.00045
d = 0.008
Δt = 30 s
desired & measured flow
damper
inlet pres.
flow to room
int. gain
m³/s or kPa or kW/m²
1.2
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
15
16
17
Time, h
s1205c
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i

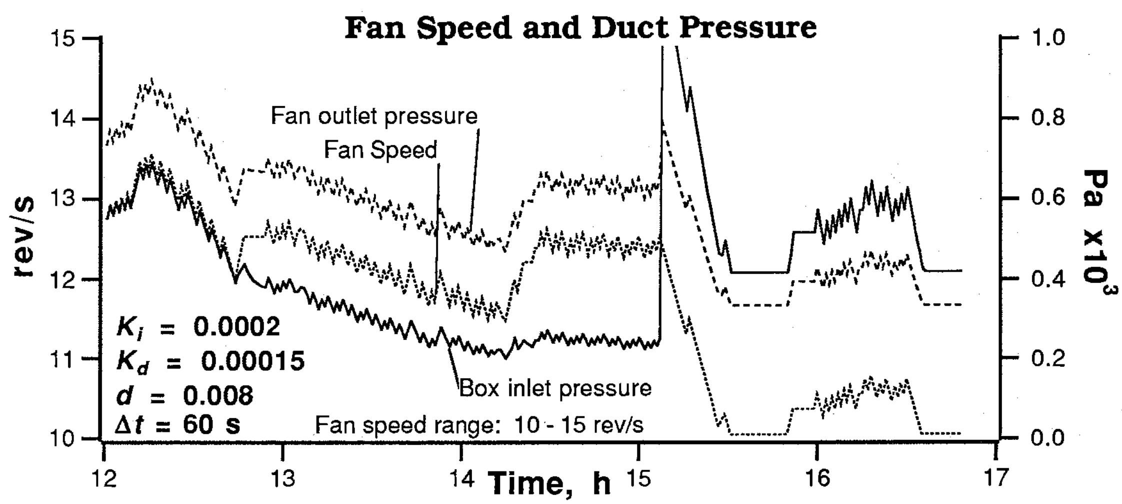


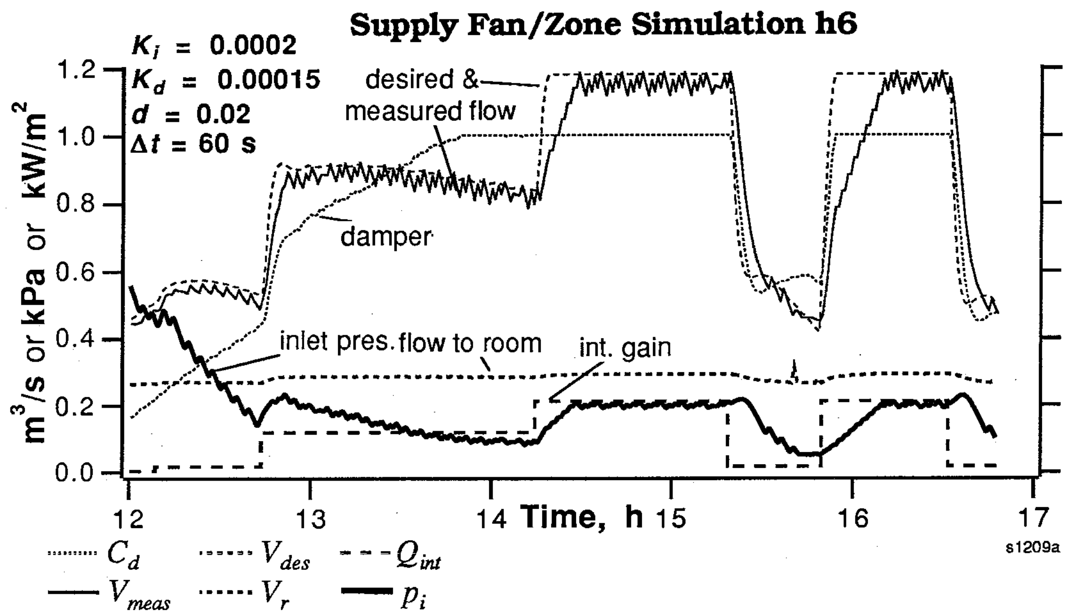
Fan Speed and Duct Pressure
K_i = 0.0003
K_d = 0.00045
d = 0.008
Δt = 30 s
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 11.25 - 15 rev/s
rev/s
15
14
13
12
11
10
Pa x10³
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
15
16
17
Time, h

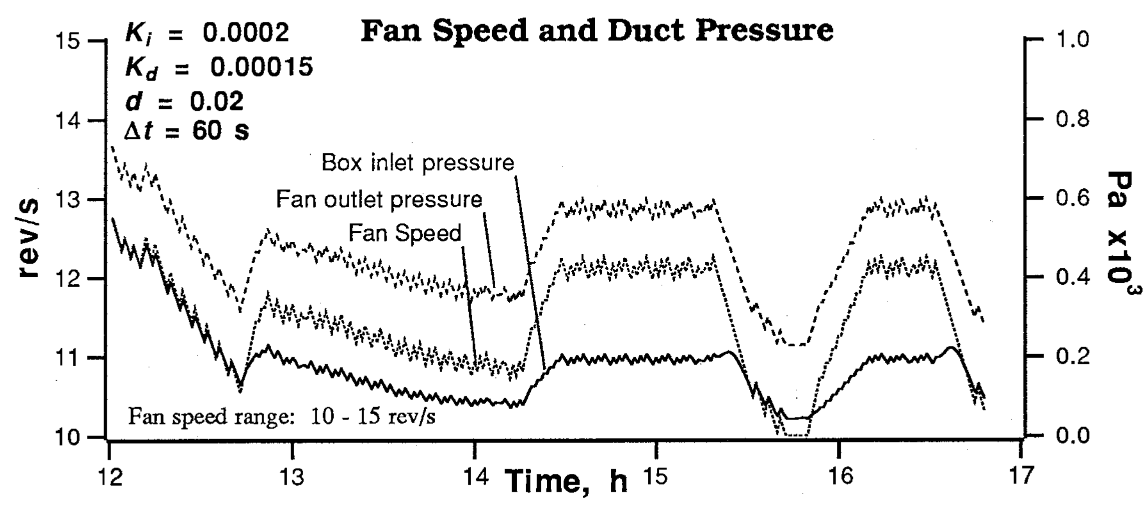

Supply Fan/Zone Simulation h4
K_i = 0.00045
K_d = 0.00045
d = 0.008
Δt = 30 s
desired & measured flow
inlet pres.
damper
flow to room
int. gain
m^3/s or kPa or kW/m^2
Time, h
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i
s1208a

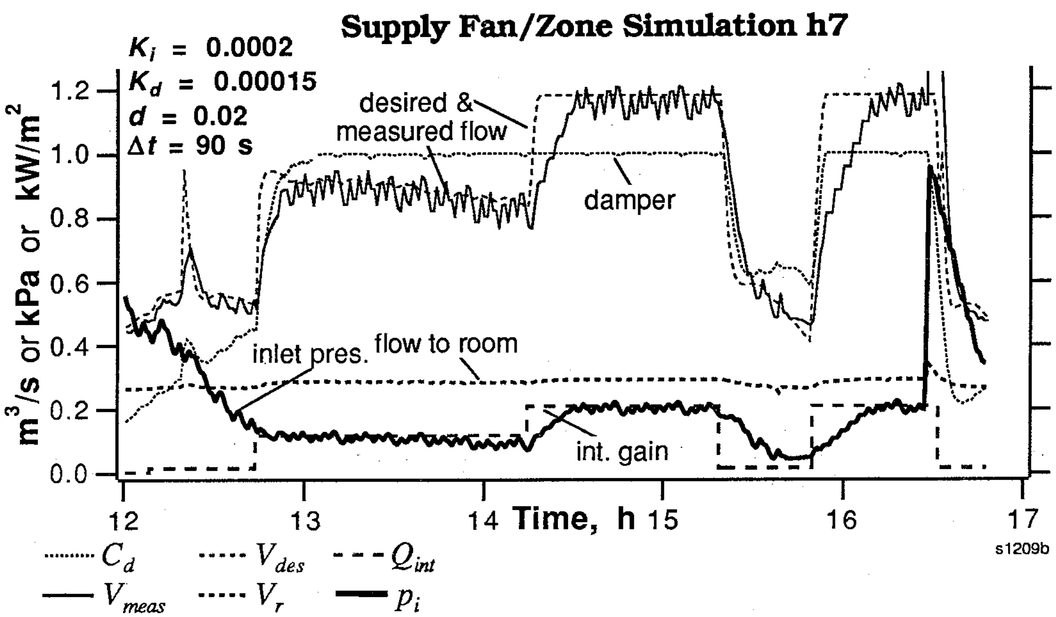

Fan Speed and Duct Pressure
Fan outlet pressure
Fan Speed
Box inlet pressure
K_i = 0.00045
K_d = 0.00045
d = 0.008
Δt = 30 s
Fan speed range: 10 - 15 rev/s
rev/s
Pa x10^3
Time, h

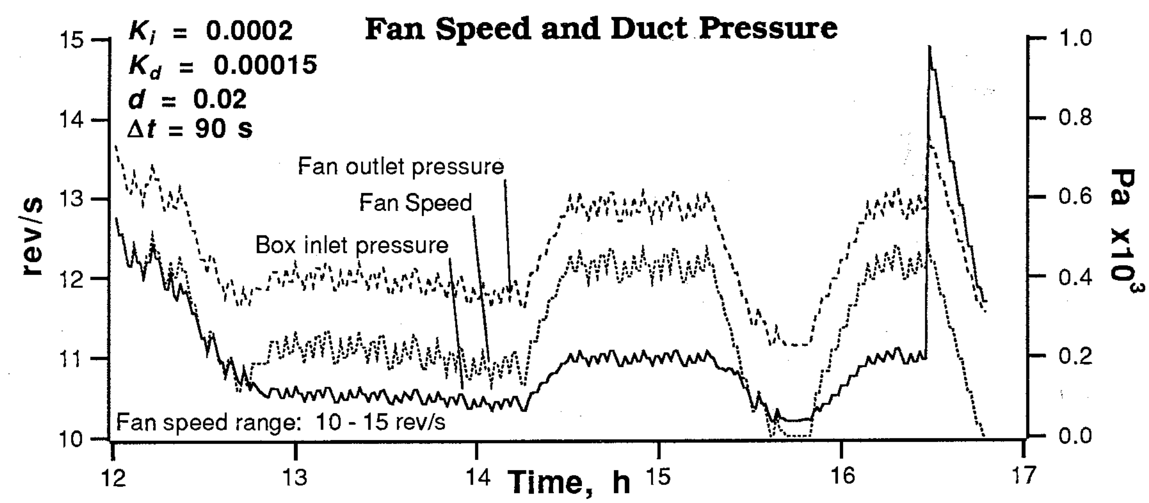

Supply Fan/Zone Simulation h5
K_i = 0.0002
K_d = 0.00015
d = 0.008
Δt = 60 s
desired & measured flow
damper
inlet pres.
flow to room
int. gain
m³/s or kPa or kW/m²
1.2
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i
s1211b

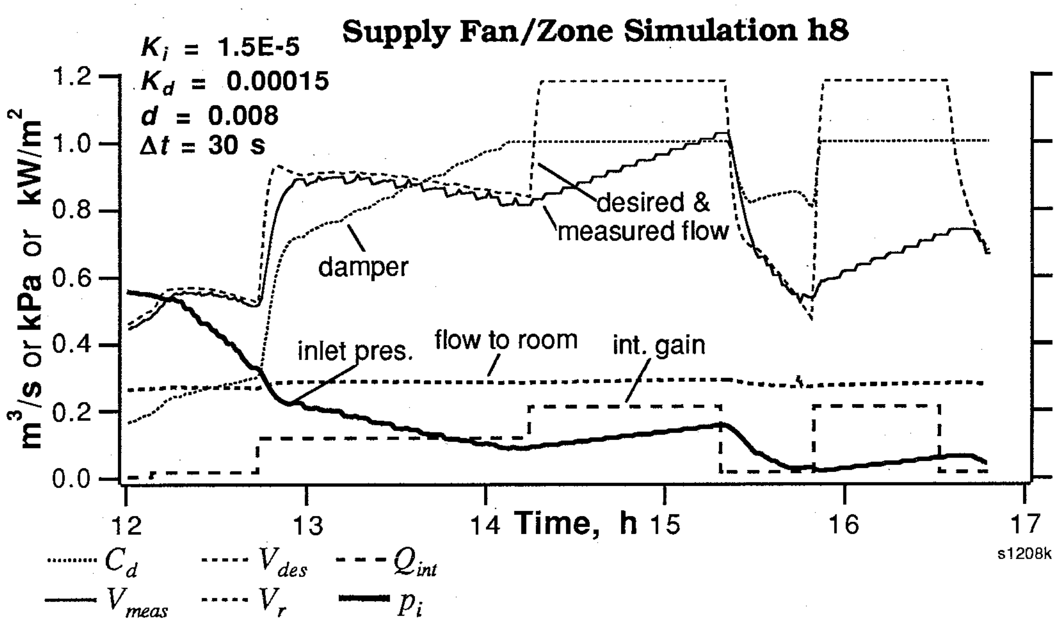

Fan Speed and Duct Pressure
rev/s
15
14
13
12
11
10
Fan outlet pressure
Fan Speed
Box inlet pressure
K_i = 0.0002
K_d = 0.00015
d = 0.008
Δt = 60 s
Fan speed range: 10 - 15 rev/s
Pa x10³
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17

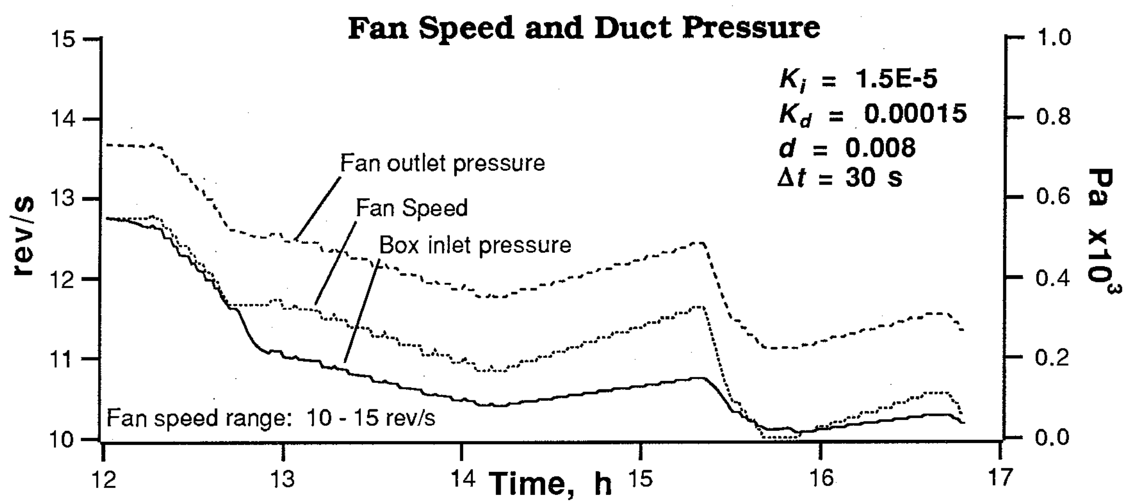

Supply Fan/Zone Simulation h6
K_i = 0.0002
K_d = 0.00015
d = 0.02
Δt = 60 s
desired & measured flow
damper
inlet pres.
flow to room
int. gain
m^3/s or kPa or kW/m^2
1.2
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17
s1209a
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i

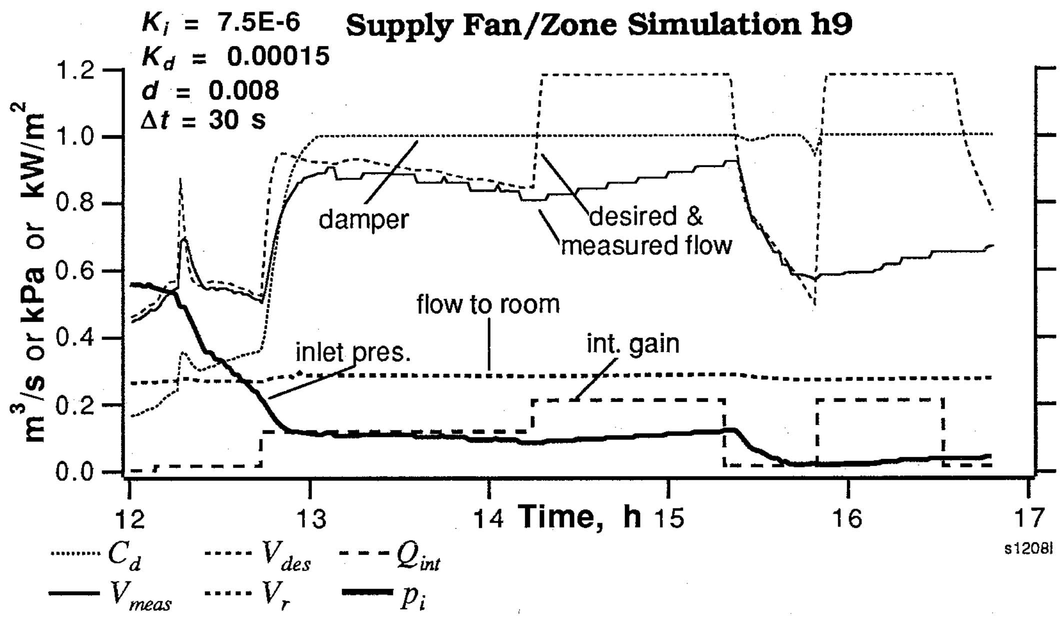

Fan Speed and Duct Pressure
K_i = 0.0002
K_d = 0.00015
d = 0.02
Δt = 60 s
Box inlet pressure
Fan outlet pressure
Fan Speed
Fan speed range: 10 - 15 rev/s
rev/s
Pa x10^3
15
14
13
12
11
10
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17

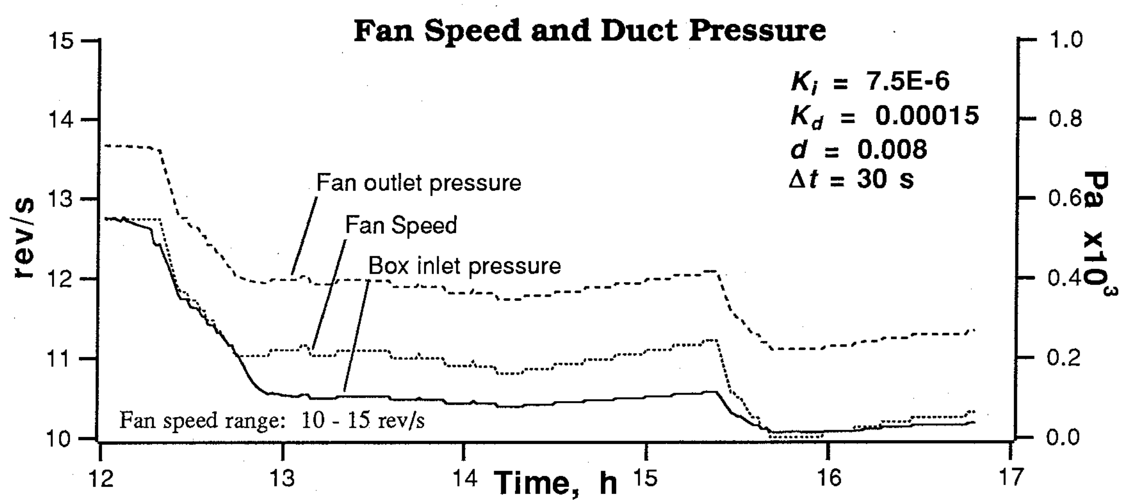

Supply Fan/Zone Simulation h7
K_i = 0.0002
K_d = 0.00015
d = 0.02
Δt = 90 s
desired &
measured flow
damper
inlet pres.
flow to room
int. gain
m³/s or kPa or kW/m²
1.2
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17
s1209b
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i

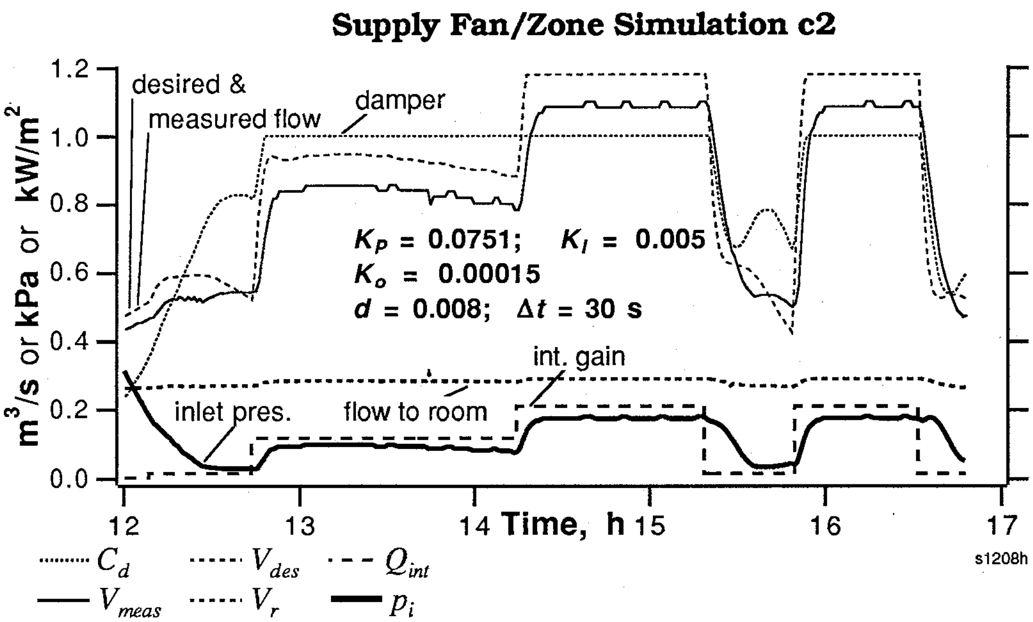

Fan Speed and Duct Pressure
K_i = 0.0002
K_d = 0.00015
d = 0.02
Δt = 90 s
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
rev/s
15
14
13
12
11
10
Pa x10³
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17

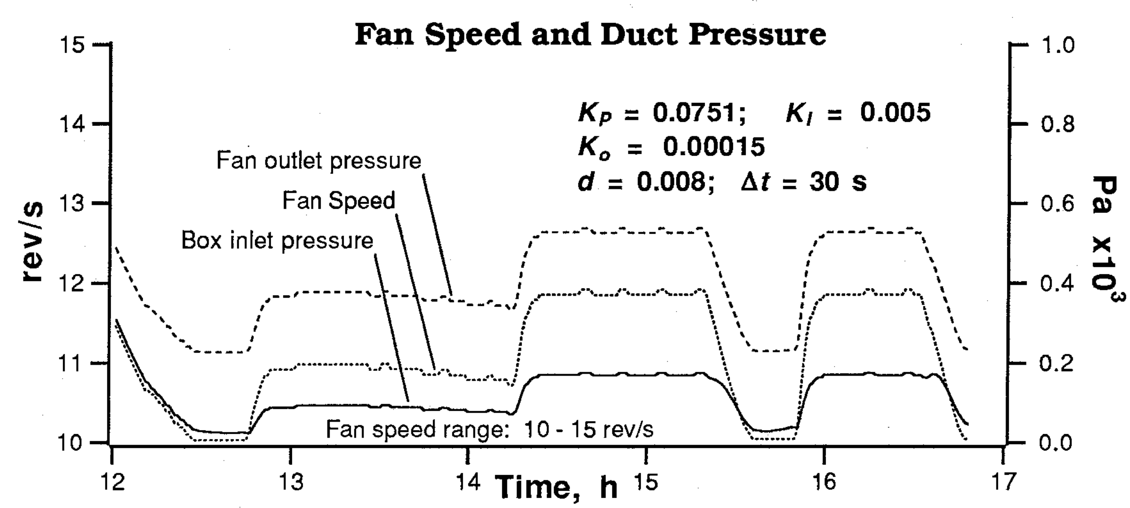

Supply Fan/Zone Simulation h8
K_i = 1.5E-5
K_d = 0.00015
d = 0.008
Δt = 30 s
m³/s or kPa or kW/m²
desired & measured flow
damper
inlet pres.
flow to room
int. gain
Time, h
s1208k
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i

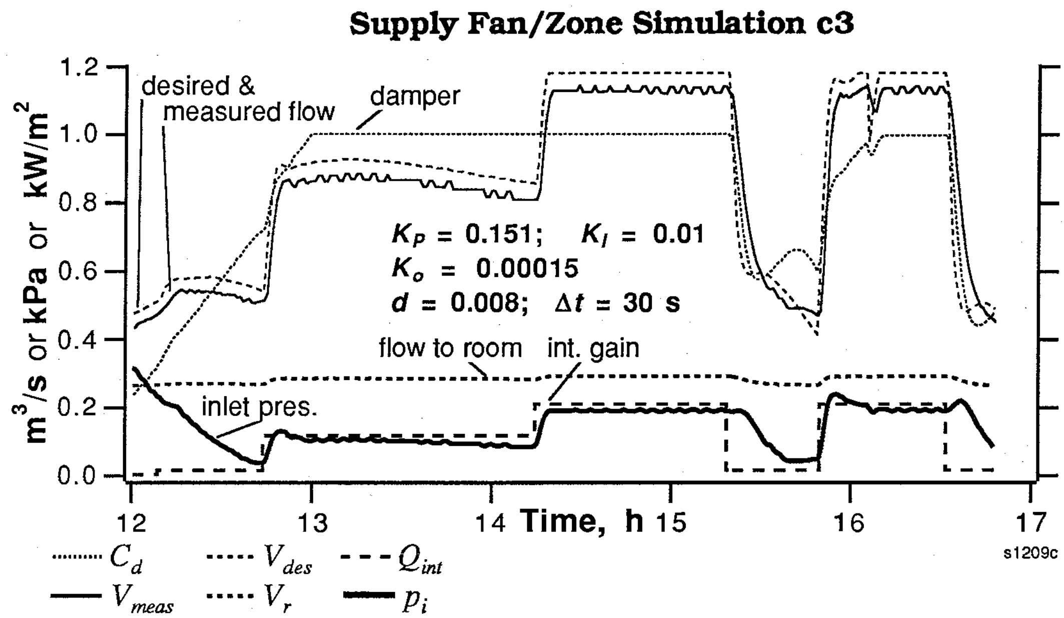

Fan Speed and Duct Pressure
K_i = 1.5E-5
K_d = 0.00015
d = 0.008
Δt = 30 s
rev/s
Pa x10³
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
Time, h

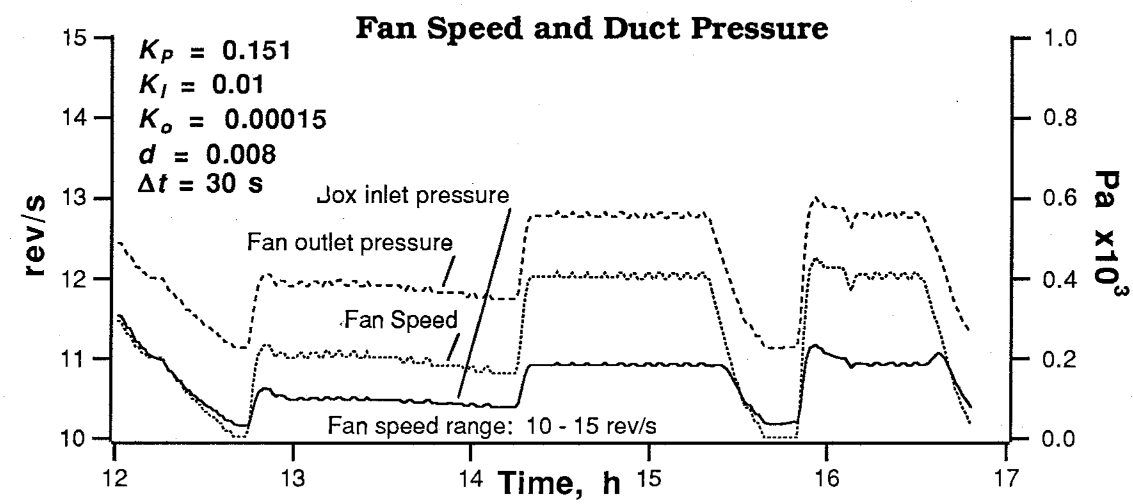

Supply Fan/Zone Simulation h9
K_i = 7.5E-6
K_d = 0.00015
d = 0.008
Δt = 30 s
m^3/s or kPa or kW/m^2
1.2
1.0
0.8
0.6
0.4
0.2
0.0
damper
desired &
measured flow
flow to room
inlet pres.
int. gain
12
13
14
Time, h
15
16
17
s1208l
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i

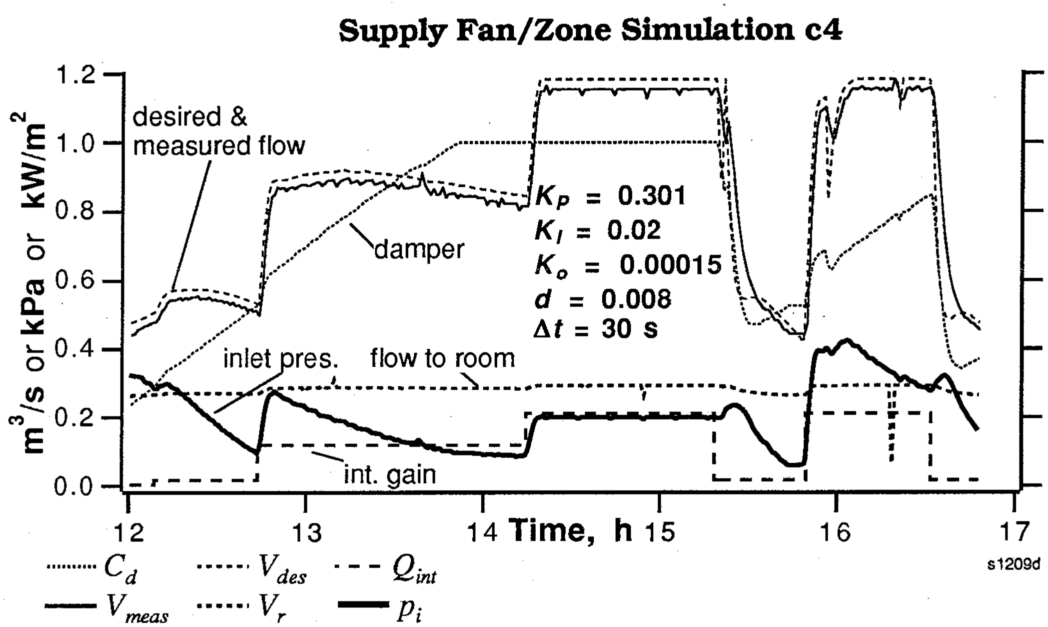

Fan Speed and Duct Pressure
K_i = 7.5E-6
K_d = 0.00015
d = 0.008
Δt = 30 s
rev/s
15
14
13
12
11
10
Pa x10^3
1.0
0.8
0.6
0.4
0.2
0.0
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
12
13
14
Time, h
15
16
17

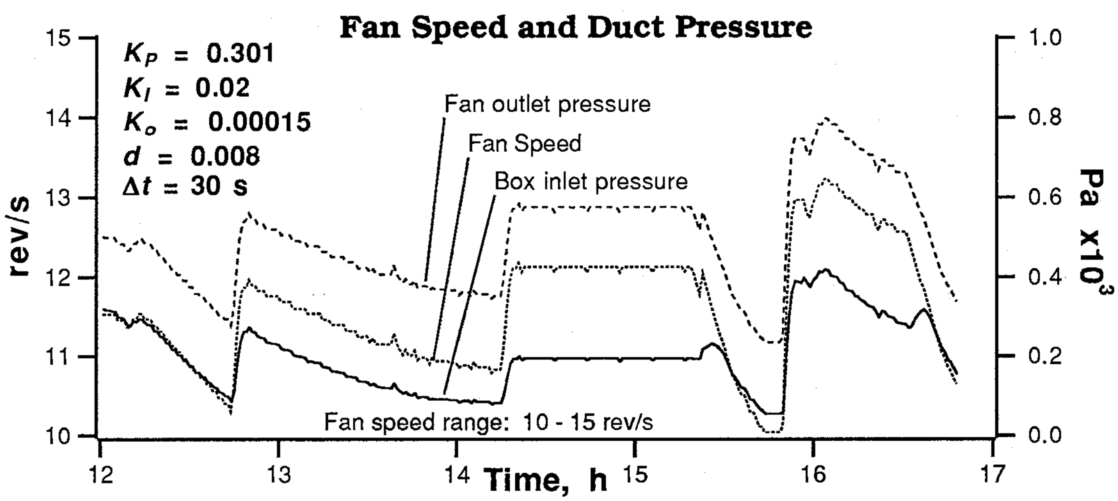

Supply Fan/Zone Simulation c2
m³/s or kPa or kW/m²
1.2
1.0
0.8
0.6
0.4
0.2
0.0
desired &
measured flow
damper
Kp = 0.0751; KI = 0.005
Ko = 0.00015
d = 0.008; Δt = 30 s
int. gain
inlet pres.
flow to room
12
13
14 Time, h 15
16
17
Cd
Vdes
Qint
Vmeas
Vr
pi
s1208h

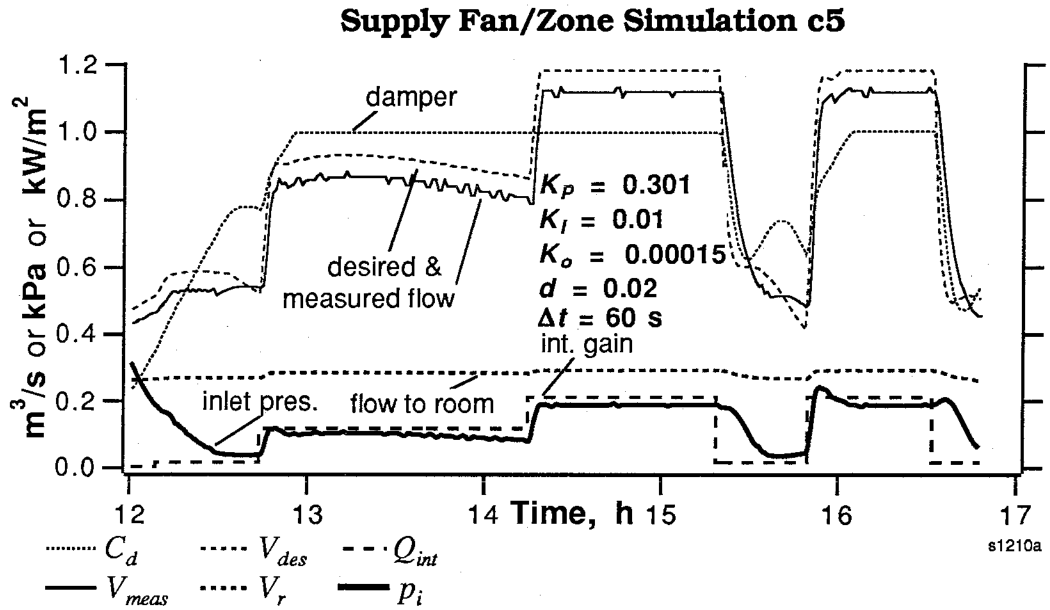

Fan Speed and Duct Pressure
rev/s
15
14
13
12
11
10
Pa x10³
1.0
0.8
0.6
0.4
0.2
0.0
Kp = 0.0751; KI = 0.005
Ko = 0.00015
d = 0.008; Δt = 30 s
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
12
13
14 Time, h 15
16
17

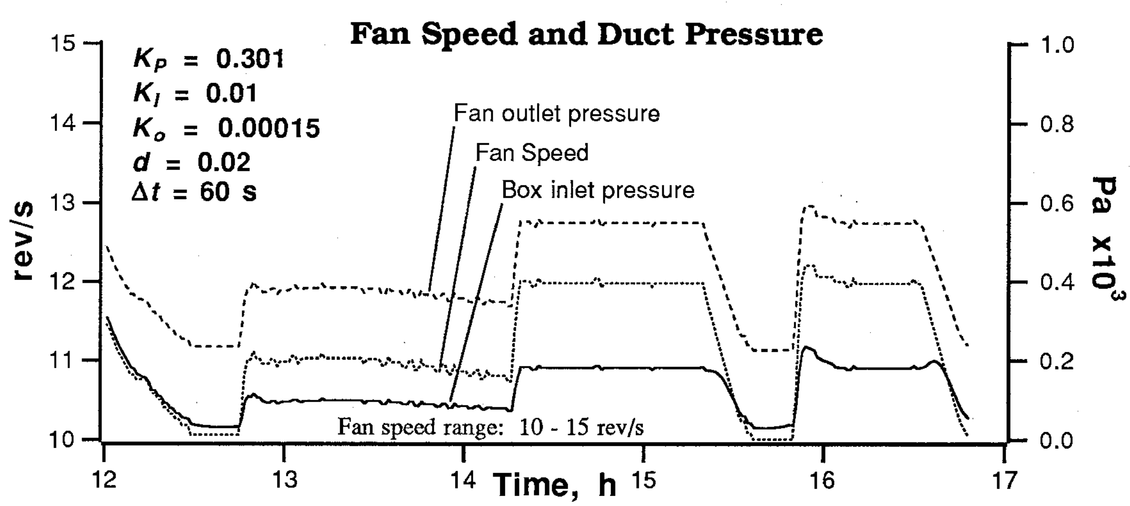

Supply Fan/Zone Simulation c3
desired &
measured flow
damper
K_P = 0.151; K_I = 0.01
K_o = 0.00015
d = 0.008; Δt = 30 s
flow to room
int. gain
inlet pres.
m³/s or kPa or kW/m²
1.2
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17
s1209c
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i

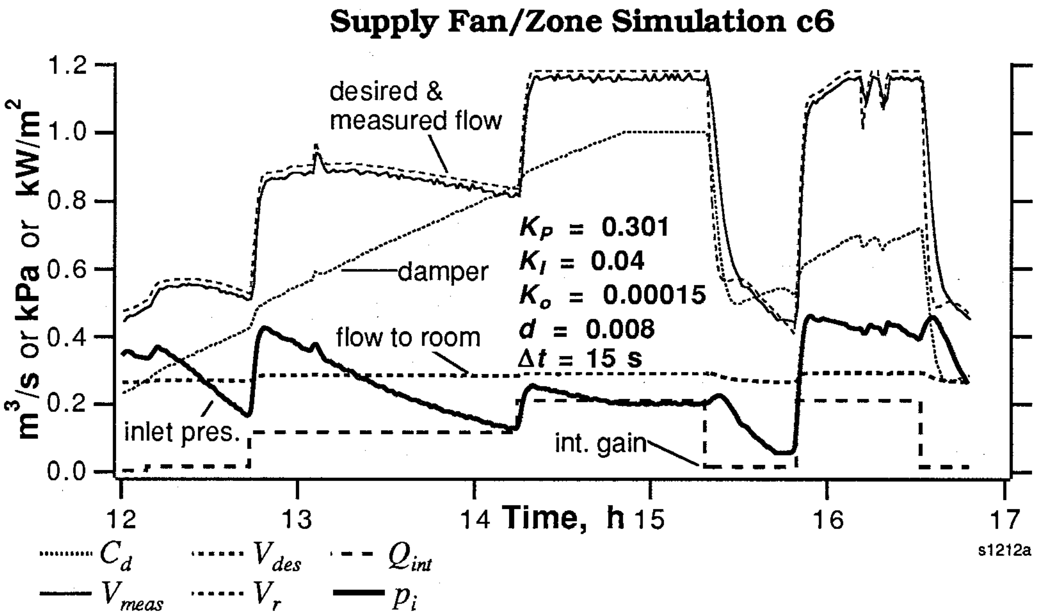

Fan Speed and Duct Pressure
K_P = 0.151
K_I = 0.01
K_o = 0.00015
d = 0.008
Δt = 30 s
Jox inlet pressure
Fan outlet pressure
Fan Speed
Fan speed range: 10 - 15 rev/s
rev/s
15
14
13
12
11
10
Pa x10³
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17

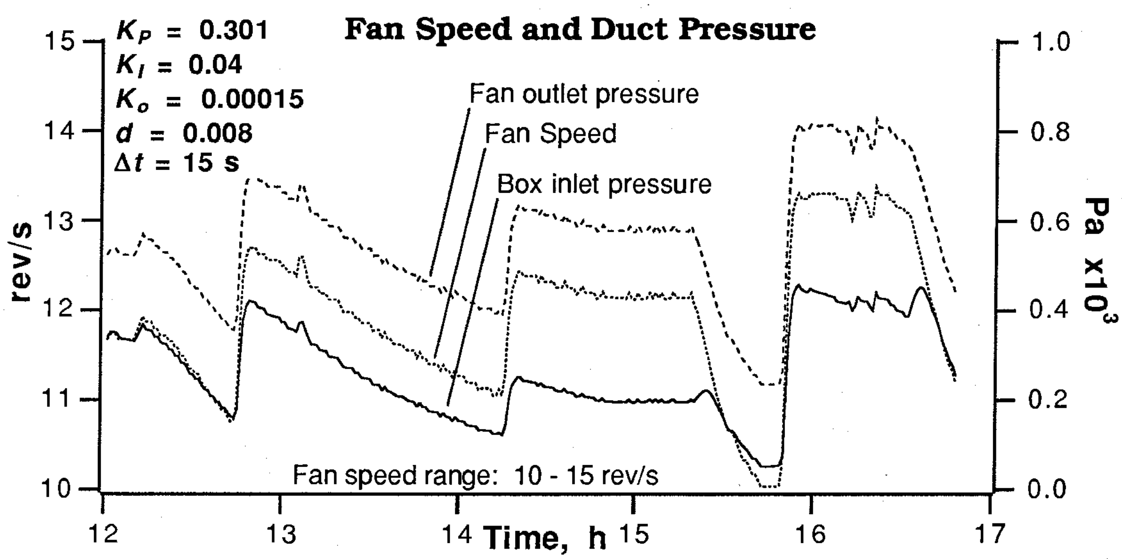

Supply Fan/Zone Simulation c4
desired & measured flow
damper
inlet pres.
flow to room
int. gain
K_P = 0.301
K_I = 0.02
K_o = 0.00015
d = 0.008
Δt = 30 s
m³/s or kPa or kW/m²
1.2
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17
s1209d
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i

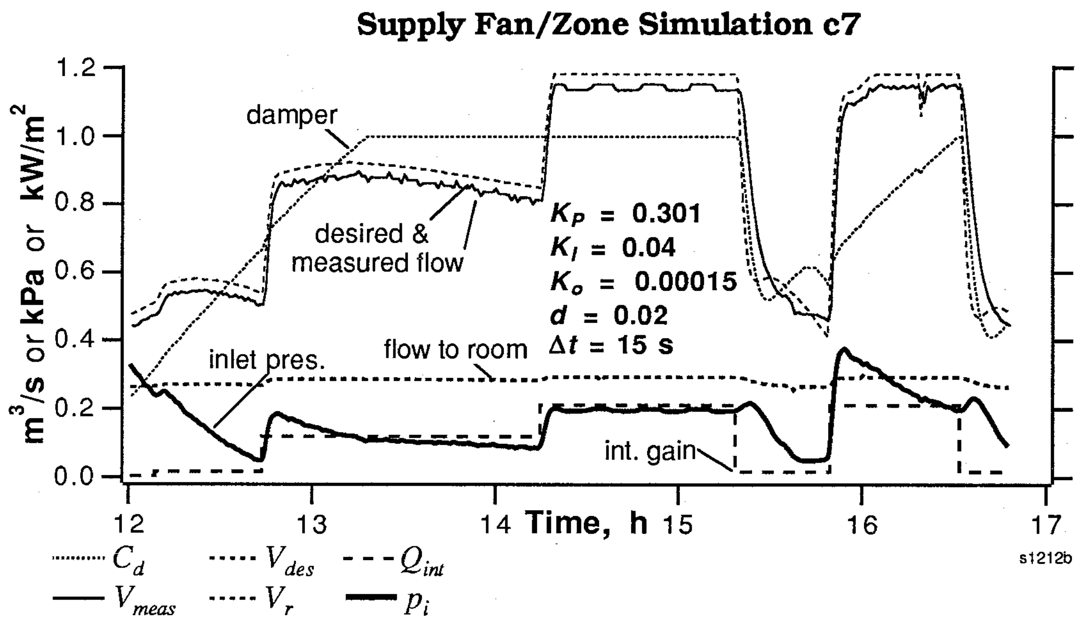

Fan Speed and Duct Pressure
K_P = 0.301
K_I = 0.02
K_o = 0.00015
d = 0.008
Δt = 30 s
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
rev/s
15
14
13
12
11
10
Pa x10³
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17

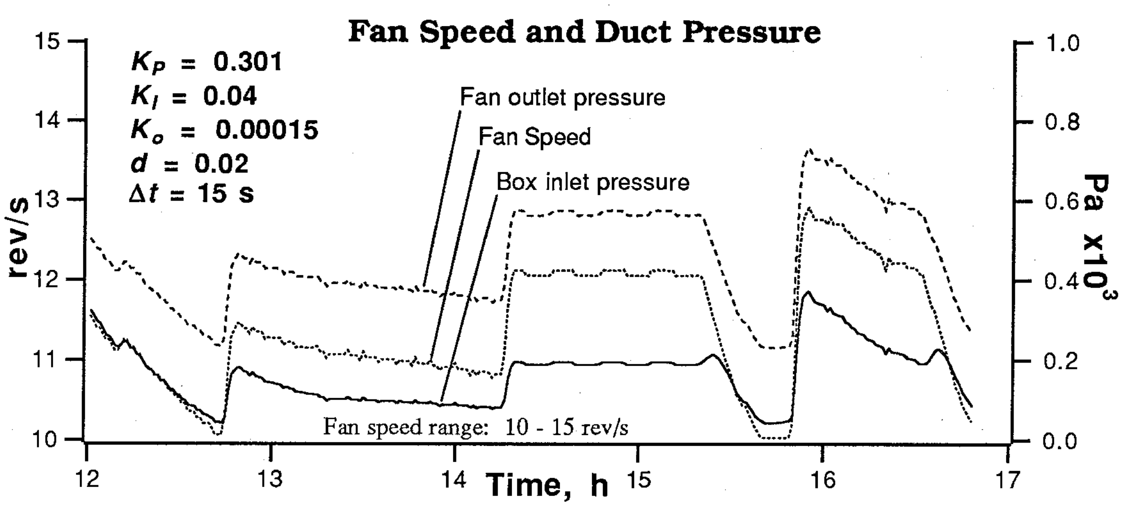

Supply Fan/Zone Simulation c5
m³/s or kPa or kW/m²
damper
desired &
measured flow
Kp = 0.301
KI = 0.01
Ko = 0.00015
d = 0.02
Δt = 60 s
int. gain
inlet pres.
flow to room
Time, h
Cd
Vdes
Qint
Vmeas
Vr
pi
s1210a

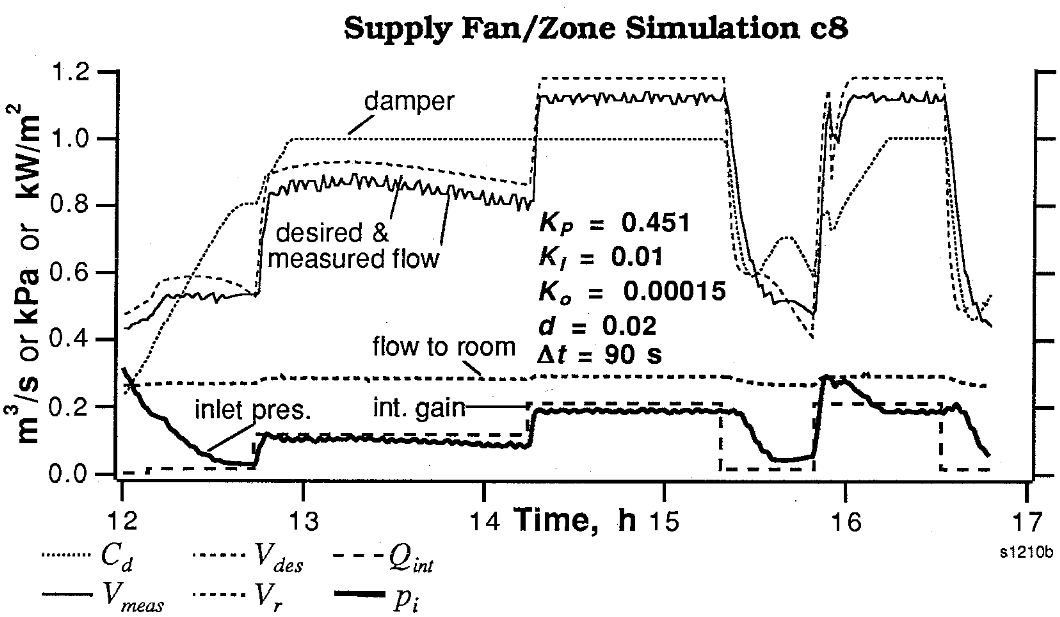

Fan Speed and Duct Pressure
rev/s
Pa x10³
Kp = 0.301
KI = 0.01
Ko = 0.00015
d = 0.02
Δt = 60 s
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
Time, h


Supply Fan/Zone Simulation c6
m³/s or kPa or kW/m²
desired & measured flow
damper
flow to room
inlet pres.
int. gain
Kp = 0.301
KI = 0.04
Ko = 0.00015
d = 0.008
Δt = 15 s
Time, h
s1212a
Cd
Vdes
Qint
Vmeas
Vr
pi

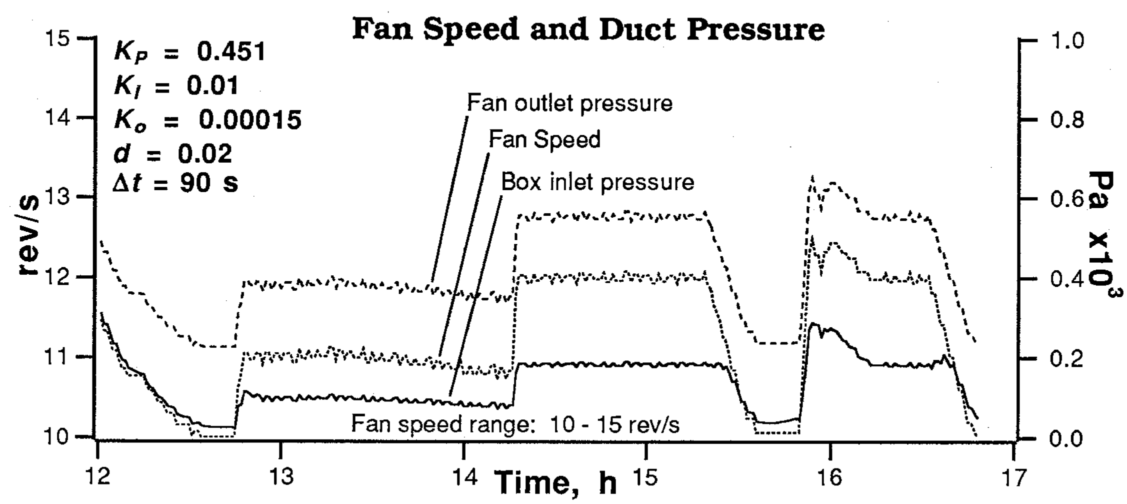

Fan Speed and Duct Pressure
Kp = 0.301
KI = 0.04
Ko = 0.00015
d = 0.008
Δt = 15 s
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
rev/s
Pa x10³
Time, h


Supply Fan/Zone Simulation c7
damper
desired &
measured flow
K_P = 0.301
K_I = 0.04
K_o = 0.00015
d = 0.02
Δt = 15 s
inlet pres.
flow to room
int. gain
m³/s or kPa or kW/m²
1.2
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17
s1212b
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i

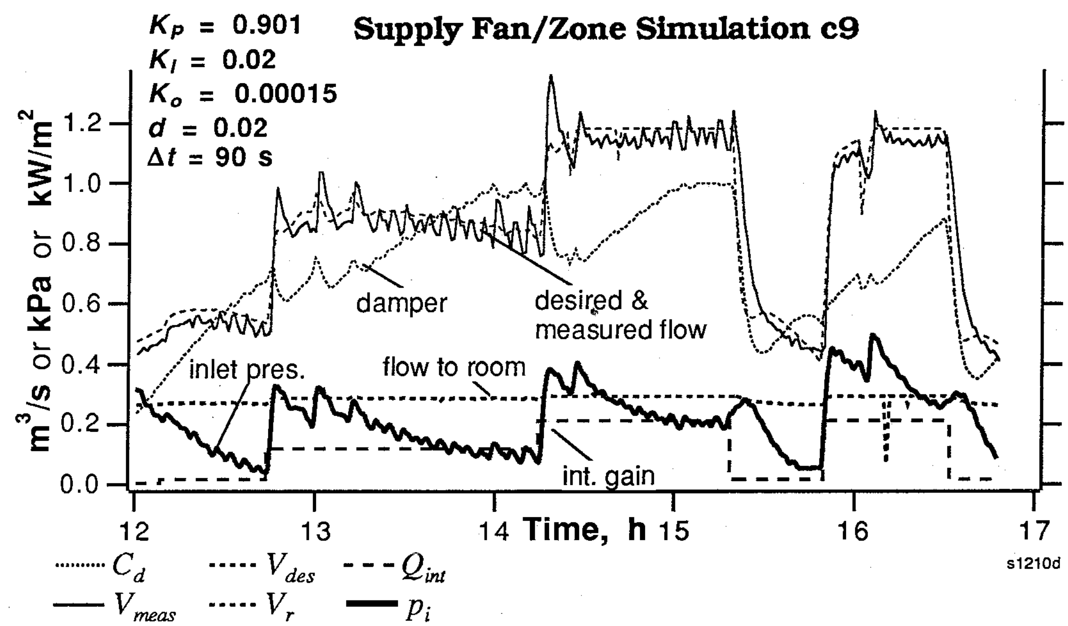

Fan Speed and Duct Pressure
K_P = 0.301
K_I = 0.04
K_o = 0.00015
d = 0.02
Δt = 15 s
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
rev/s
15
14
13
12
11
10
Pa x10³
1.0
0.8
0.6
0.4
0.2
0.0
12
13
14
Time, h
15
16
17



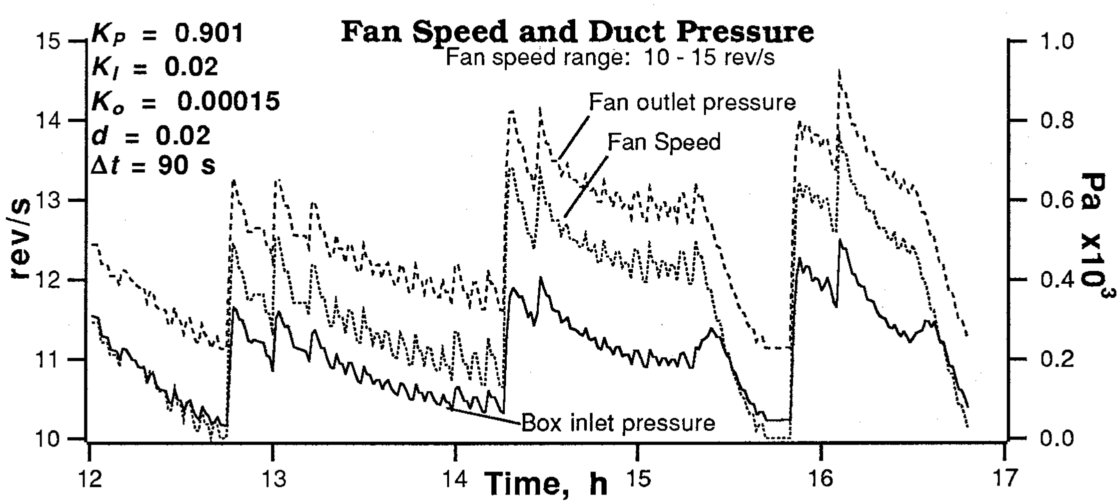
Supply Fan/Zone Simulation c8
damper
desired &
measured flow
K_P = 0.451
K_I = 0.01
K_o = 0.00015
d = 0.02
Δt = 90 s
flow to room
inlet pres.
int. gain
m^3/s or kPa or kW/m^2
Time, h
s1210b
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i



Fan Speed and Duct Pressure
K_P = 0.451
K_I = 0.01
K_o = 0.00015
d = 0.02
Δt = 90 s
Fan outlet pressure
Fan Speed
Box inlet pressure
Fan speed range: 10 - 15 rev/s
rev/s
Pa x10^3
Time, h



Supply Fan/Zone Simulation c9
K_P = 0.901
K_I = 0.02
K_o = 0.00015
d = 0.02
Δt = 90 s
m^3/s or kPa or kW/m^2
damper
desired & measured flow
inlet pres.
flow to room
int. gain
Time, h
C_d
V_{des}
Q_{int}
V_{meas}
V_r
p_i
s1210d



Fan Speed and Duct Pressure
Fan speed range: 10 - 15 rev/s
K_P = 0.901
K_I = 0.02
K_o = 0.00015
d = 0.02
Δt = 90 s
Fan outlet pressure
Fan Speed
Box inlet pressure
rev/s
Pa $x10^3$
Time, h

Chapter 5

Discussion: Modular HVAC Dynamic Simulation

5.1 Problems with *HVACSIM+* and the VAV System Model

To its credit, *HVACSIM+* proved to be a powerful tool for testing new digital control techniques. Getting it compiled and running on an operating system and hardware unlike that used by its developers required relatively minor modifications and only a basic knowledge of Fortran and the local compilers and linkers. Because the program is in the public domain, using it involves no capital investment in software, assuming one already has access to a compiler. The documentation is generally clear, and includes good examples to help the novice user get started. The *HVACSIM+* library contains most of the components required to build a model of a typical piece of HVAC equipment, and it is possible to write custom component models.

As useful as *HVACSIM+* was for the simulations performed here, the time and effort involved in coping with its shortcomings was considerable. A few of these weaknesses are highlighted here:[1]

[1]HVACSIM+ actually consists of three separate programs: a model generation program, a model "compiler" and the simulation program itself. In the following discussion, references to HVACSIM+ imply the complete package..

Error detection. Often errors in model specification on the part of the user were to blame for simulation failure, however the the inability of the program to prevent or detect such errors exacerbated this problem. Although the program does not allow the user to commit certain mistakes, such as connecting a pressure output to a temperature input, and will detect whether two or more outputs are identified as a single state variable, it cannot recognize an underdetermined model[2] or similar breeches of rules fundamental to sound model definition, and permits the use of initial values that might result in division by zero.

User interface. Probably for the sake of portability, *HVACSIM+* uses a text-based user interface, i.e., all input from the user is typed in response to questions typed by the program, and all simulation results are typed a line at a time on the screen or written to a file. Creating or editing a model is fairly tedious, requiring the user to navigate (by keystroke response to a prompt) a hierarchy of menus—crude by today's standards for microcomputer or workstation software, but justifiable, as this feature enables a single version of source code to run on a broad range of machines, from PCs to Crays. Editing an existing model involves entering this environment to make a change, then recompiling the simulation into "executable" form. Thus creation of large models is time-consuming, and rapid modification of model parameters is impossible.[3]

Likewise, all output is in text form. Model definitions can be viewed by listing the state variables corresponding to each component's inputs and outputs.[4] Is is virtually a necessity for the user to work from a model diagram such as those presented in Chapter 3, in order to avoid confusion. Simulation results consist of a sequence of reported variable values for each reporting interval. Data are written to an output file in a non-standard

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[2]The novice can easily produce an underdetermined model by omitting a key component such as an inlet conduit, in which flow rate is calculated from inlet and outlet pressures; flow rate is an input for most other components.
[3]An HVACSIM+ "hacker" will resort to directly editing the executable files in order to avoid this process, for making certain parameter changes. This is not recommended.
[4]Several examples are given in the listings in Appendix C.

form that requires further processing into a "row and column" (one record per interval) form readable by most analysis or plotting programs.[5] A "summary" file provides variable name headings at each interval for easier preliminary interpretation of results, but is limited in the number of reporting intervals it records. Plotting of results is left to the user.

Lack of inherent "intelligence." Beyond requiring the user to specify the proper type of variable for a given component input or output, *HVACSIM+* does little to enforce the rules on which it is based or use them to facilitate model building:

- In creating a simulation, the user must specify the particular variables that correspond to the inputs and outputs of each component, a process vulnerable to error. Liu (& Kelly, 1988) has developed an improved text-based interface that employs rule-based methods to do this automatically; the user simply specifies the physical relationship of components. This effort is only a prototype, however, and does not support all the default library components; it does not allow for the addition of user-created types. The text-based nature of the interface remains a major constraint.

- A number of user-specified parameters influence the behavior of the equation solver: a convergence parameter, tolerances used to determine whether a variable should be frozen or unfrozen, a time interval over which the integral of a differential equation must be within these tolerances, superblock freezing options, and minimum and maximum time steps. Poor selection of parameter values can lead to accuracy problems so severe that convergence is threatened, as observed in some of the simulations presented here. The user manual gives only cursory guidance on the choice of values;[6] trial and error selection is complicated by their interaction. Haves and Trewhella (1988) have done preliminary work to investigate the effect of these parameters on accuracy and execution, but conclude that more work is needed to develop a set of rules for parameter selection.

Computational problems. In the course of the simulations performed in this study, occasionally slight (i.e., non-fatal) non-convergence was observed. Reducing the time step, decreasing the tolerances, making the freezing options more stringent, or modifying boundary data values usually corrected problems of this magnitude. As discussed

[5]This post-processing may be done simultaneously in multitasking environments such as UNIX. The Center for Energy and Environmental Studies' *Archive* software (Feuermann and Kempton, 1987) proved invaluable for processing the results of simulations with more than one superblock, as well as for performing unit conversions.

[6]The program does enforce one rule relating the tolerances that is given in the manual.

in Chapter 4, several configurations of the supply distribution duct system were not able to be simulated at all. Non-convergence led to floating point overflow errors for no apparent reason; in one case decoupling the temperature equations provided a work-around. Large (i.e., many components) duct networks with splits and merges seem to be prone to this sort of difficulty. This sort of problem is frustrating to the user, as neither the exact cause nor an appropriate solution is evident; finding a solution is time-consuming if not impossible. The problem may not manifest itself until late in a long-running simulation, thus limiting the speed with which potential remedies can be found.

Problems caused by discontinuities in time-varying boundary conditions are generally easier to track down and fix than others—a time series plot of the non-convergent variables and the boundary variables will reveal a causal relationship. When *HVACSIM+* encounters an abrupt change in a boundary variable, it must reset the differential equation integration algorithm, but must do this one time step in advance. Normally, the user indicates the approach of a discontinuity by including two data records with the same time value. In the simulations performed here, sometimes this did not work—in the case of very large discontinuities, as for internal heat gain—or was not practical, as with using measured data (containing many discontinuities) for boundary values. In the former case, it was found that reducing the boundary data time step[7] by an order of magnitude for one time step, resulting in ten smaller steps, and simulating the step change with a ramp over these ten steps, with a reset before and after, was successful. The problem of discontinuous real data was solved easily by smoothing the data.

Speed. Five hour simulations of the most complex model used in this study (described in Chapter 4) took longer than a day to run, with *HVACSIM+* running on a Sun 3/60 workstation, with MC68881 floating point processor installed.[8] Testing the influence of model parameter changes was thus a time-consuming process.

[7]Much larger, here, than the simulation time step.
[8]With about 90% of the CPU time devoted to the program.

Technical support. Although the developers of *HVACSIM+* at NIST are willing to answer users' questions and offer suggestions, the effort they are able to put into this is limited, as there is currently no funding for technical support of the software. There are several users (internationally) with a good deal of experience who can offer advice and share component models, but there is no organized group. Although *TRNSYS*[9] may not be an appropriate tool for certain control simulations, it has a large user base, and is well supported by the developer.

5.2 New Directions in Dynamic HVAC Simulation

Graphical/Intelligent User Interfaces

In general, problems of error detection, user interface, and intelligence are not peculiar to *HVACSIM+,* and are being addressed by several groups working on the next generation of dynamic building simulation programs. One such package, *ESP*, developed by Joe Clarke at the Energy Simulation Research Unit of the University of Strathclyde (Clarke and McClean, 1988), has as its core a simulation engine and building component library similar to that of *HVACSIM+* and *TRNSYS*, integrated with extensive graphical input/output capabilities, and implemented on UNIX workstations. A notable feature of the system is an "expert" mode, which can perform standard performance appraisals such as comfort assessment, plant sizing, control system analysis, and energy consumption estimation, using modifiable rules scripts.

Other groups are working on graphical "front ends" for existing programs: Haves and Trewhella (1988) at the University of Oxford are developing a system for UNIX workstations linking the CES package (developed at the University of Wales at Swansea) which includes block diagram, icon, and signal flow graph editors to *HVACSIM+,* as well as adding real-time graphical output; a team at the Solar Energy Research Center at the

[9]The program developed at the Solar Energy Laboratory at the University of Wisconsin, discussed in Chapter 3. This program is not in the public domain; it is distributed on a licensed basis by the Laboratory.

University College of Falun/Borlänge (Sweden) and the Monitoring Center for Energy Research at the Royal Institute of Technology in Stockholm are developing a CAD-style model generation pre-processor for microcomputers running DOS, intended for use with *TRNSYS* and other modular HVAC simulation programs, and a similar program for the Macintosh is being developed by the Solar Energy Laboratory at the University of Wisconsin (Solar Energy Laboratory, 1989).

The availability of standard graphics-based environments (such as the *X-Window System*) that run on a broad range of machines has increased the portability of graphical user interfaces, thereby lessening the programming effort required to enable a single version of simulation software to run on several machines.

A NEW EQUATION SOLVING APPROACH

A new approach to solving non-linear systems is used in the Simulation Problem Analysis Kernel (SPANK). SPANK is a general object-oriented simulation environment developed at Lawrence Berkeley Laboratory, which uses a directed graph rather than a matrix as the primary data structure, thus exploiting the topology of the model to reduce the number of variables active in each iteration of the equation solver (Sowell et al., 1986). The program, still under development, has been tested with steady-state and dynamic building and HVAC plant models. Haves (1987) compared SPANK with *HVACSIM+*, and reported that for models consisting of many simple components in series (typical of the duct systems modeled in this study), speed improvements of an order of magnitude may be possible. Such an advance could greatly enhance the application of dynamic simulation to design problems, as execution speed is one of the main factors currently limiting its primary use to research.

REAL-TIME APPLICATION OF SIMULATION TOOLS

A thorough treatment of this topic cannot be given here, however an important development related to some of the issues dealt with in this thesis bears mention: emulation and testing of building energy management systems (BEMSs). BEMSs have become popular over the last 15 years as retrofits or as integral parts of control systems in new construction. The success of BEMS installations, from an energy savings perspective, has been mixed, leading building owners and managers to take a justifiably skeptical approach to investment in such systems, as there has been no feasible way to predict or guarantee the benefits.[10] Controlled evaluation of BEMSs already installed in buildings is difficult, because it impossible to run tests under identical conditions; engineering analysis of the control methods used is limited, as the algorithms are usually proprietary.

At least two groups (Haves and Dexter, 1989; Kelly and May, 1990) are pioneering the use of systems which can emulate a building and its mechanical and control systems via simulation, for testing the performance of BEMSs. Emulator/testers use a computer to simulate a building or building system, and connect to the BEMS with a hardware interface. Both groups mentioned here use *HVACSIM+* for simulation; Haves' system consists of a workstation running the simulation, connected via serial link to the interface. The interface, which connects the analog and digital inputs and outputs of the BEMS to the sensors and actuators of the simulated plant, is a network of microcomputers (one for each station of the BEMS) with peripheral boards to perform the analog to digital, digital to analog, and digital to digital[11] conversion and multiplexing of the control signals. Communication between the simulation and the interface is controlled by a component model which initiates data transfers. Kelly and May have proposed preliminary BEMS

[10] There are firms that offer guaranteed or shared savings programs for the installation of a BEMS, often as part of a larger energy conservation package.

[11] i.e., between digital control signals at BEMS logic levels and data transmitted to and from the workstation.

rating criteria, weather conditions, building/HVAC types, and building use conditions for standard evaluation of BEMSs.

5.3 Conclusion

Recent advances made by researchers developing intelligent front ends for dynamic HVAC simulation programs will soon make it possible to use these tools feasibly in a design environment, and enhance their use in research. Improved equation solving techniques being incorporated into simulation tools will likewise contribute to this transformation. Graphical user interfaces to dynamic simulation tools will be truly useful for HVAC control design when the user can, on screen, slide a button icon representing controller gain or other parameter, and watch its effect on the controlled variable plotted simultaneously in another window, either in the time or frequency domain.

The realization of systems capable of evaluating BEMS performance, and the accompanying establishment of performance standards for this equipment will no doubt increase the quality and energy saving potential of BEMS products on the market, and increase their utilization in buildings.

Finally, it is evident from the survey of research presented here that many of the new developments in this area are coming from Europe, while relatively little is being done in the U.S. According to one researcher (Kelly, 1990), European groups are indeed excelling in the development of building or HVAC simulation tools, as there is currently no funding available in the U.S. for work in this area,[12] although private industry continues development of proprietary software for in-house use. Clearly, the problem of funding for this work must be addressed if the technical difficulties encountered in this study, currently impeding the progress of more effective efforts, are to be eliminated.

[12]...as is the case for most energy conservation research in this country. The paucity of funding for development or support of public domain building simulation software is but one example of the deleterious impact the lack of insight by the Department of Energy and other funding agencies has had on progress in this field.

Chapter 6

Summary and Conclusion

The work presented here explores the significance of a single variable—duct static pressure—to a host of technical issues surrounding energy use and comfort in buildings equipt with variable air volume systems. In an analysis of measured energy use in variable speed drives on VAV supply fans, we have pointed out that most published power characteristics for VSDs are derived from fixed systems (i.e., systems with a fixed system constant or flow resistance coefficient), and we have demonstrated how their indiscriminate application to variable systems can result in a significant underestimation of the energy used by VSDs.

Economic considerations show VSDs to be much more attractive in new construction, or in existing systems with DDC zone control, than as a retrofit in pneumatically controlled VAV systems. As the price of VSDs continues to fall due to increased production volumes and advances in microelectronics, we can expect their position to improve in all markets.

Our measurements of fan power as a function of static pressure point to static pressure minimization as a control strategy to achieve significant energy savings, making VSDs more cost-effective. We have proposed two static pressure minimization algorithms which use feedback from local zone control loops to control the supply fan, and have shown through simulation that given the proper choice of controller parameters, the supply fan can be controlled to effectively reduce duct static pressure dynamically in a system with DDC zone control without sacrificing occupant comfort. A modified PI algorithm was shown to reduce static pressure more quickly than a heuristic algorithm, and thus appears more promising. The success of this preliminary analysis opens the way for more in-depth control studies and experimentation necessary for practical implementation of static pressure minimization control.

The DDC terminal box model and component models should prove useful to others working in this field. The process of deriving the model was perhaps more valuable than the results themselves, in that a coherent formulation of the fundamental principals underlying the components was developed, bringing new insight to an understanding of the VAV system as a whole.

The recent introduction of DDC terminal boxes has made feasible, in addition to static pressure control, improvements in zone temperature control, notably the addition of an integral mode to the flow and temperature control loops, and the possibility of automatic rebalancing, made possible through software control of fan speed. In our discussion of PI temperature control and remote rebalancing, some key issues were identified; we have posed more questions than answers, however, concluding that more work must be done in each of these areas.

The focus of this study on fan energy alone was perhaps too narrow, in a building where whole-building energy use is a result of the interaction of many components. Global optimization of all the major energy-using systems, a concept familiar to process engineers, is only recently catching on in HVAC operation (Cumali, 1988). The work done here considered supply air temperature to be a constant, as it traditionally has been in VAV systems, whereas new thinking in this area recognizes the tradeoff between supply air temperature and flow in their effect on chiller and fan energy use (Norford et al., 1986). Future work continuing the fan energy studies done here must consider optimization of fan and chiller energy use together.

In the same vein, most of the emphasis has been on cooling only, partly because we anticipated cooling loads to be a factor limiting static pressure reduction, and partly because the experiments with the DDC terminal box were performed during the cooling season. Various aspects of the heating operation of DDC terminal boxes can differ from that of their pneumatic predecessors in ways that influence energy use and comfort, and should be considered in a more thorough evaluation of the technology.

Indoor air quality is a concern that has come to the forefront of building science in recent years as the connection between ventilation air quantity and health becomes more apparent; ventilation standards and technology are rapidly changing as a result (Janssen, 1989). New methods of ventilation air control will undoubtedly appear (perhaps utilizing pollutant concentration sensing for feedback), having implications for energy use as well as some of the control issues discussed here.

In the course of this work, the experience with *HVACSIM+*, a public domain HVAC simulation tool, made evident weaknesses in several aspects of the program. A look at some of the new directions HVAC simulation is taking revealed that many of these problems are being dealt with in the next generation of simulation tools. Graphical user interfaces and the application of rule-based techniques, as well as the possibility of increased simulation speed afforded by new equation solving techniques, will do much to make HVAC simulation tools more accessible and practical for designers and researchers alike.

Finally, the work described in this thesis illustrates the importance of comprehensive *in situ* measurements and analysis of equipment performance to the application of energy conservation technology; experimentation in buildings is expensive and time-consuming, but there is no other way to expose the real-world problems that can mean success or failure for a retrofit, as we have seen in the case of excessive flow due to malfunctioning terminal boxes. Experimental work in buildings and simulation go hand in hand: Simulations are pointless unless the theory underlying models has been experimentally verified to be physically realistic. Experimental exploration of new building systems technology and control strategies is extremely limited in the range of options that can feasibly be tried, given the constraints of time and budget; simulation (which can be time consuming) is perhaps more limited by the creativity of the modeler and computational power available. Simulation, through model verification, can help the researcher or designer gain a physical understanding of the systems modeled, and can guide experimentation in the choice of parameters to be measured or tests to be performed.

References

Anderson, C. (1989). Personal communication. Titus Products, Division of Philips Industries Inc., Richardson, TX.

ASHRAE (1982). *Energy Professional Development Series, Air system design and retrofit for energy/cost effectiveness.* W.J. Coad, J.B. Graham, G.J. Williams. American Society of Heating, Refrigerating, and Air Conditioning Engineers. Atlanta, Georgia.

ASHRAE (1988). *1988 ASHRAE handbook, Equipment.* American Society of Heating, Refrigerating, and Air Conditioning Engineers. Atlanta, Georgia.

ASHRAE (1989). *1989 ASHRAE Handbook, Fundamentals.* American Society of Heating, Refrigerating, and Air Conditioning Engineers. Atlanta, Georgia.

ASHRAE/AMCA (1975). *Laboratory methods of testing fans for rating.* Atlanta: American Society of Heating, Refrigerating, and Air Conditioning Engineers; and Air Moving and Conditioning Association, Inc.

Baldwin, S. F. (1988). "Energy-efficient electric motor drive systems: Electric motor fundamentals—a tutorial." PU/CEES Working Paper no. 92, Center for Energy and Environmental Studies, Princeton University, Princeton, New Jersey.

Borresen, B. A. (1981). "Thermal room models for control analysis." *ASHRAE Transactions*, v. 87, pt. 2.

Buffalo Forge Co. (1983). *Fan engineering*, 8th ed. R. Jorgensen, ed. Buffalo, NY.

Clark, D. R. (1985). "HVACSIM+ Building systems and equipment simulation program—Reference Manual." NBSIR 84-2996. U.S. Department of Commerce, National Bureau of Standards, National Engineering Laboratory, Center for Building Technology, Gaithersburg, Maryland.

Clark, D. R., and May, W. B. (1985). "HVACSIM+ Building systems and equipment simulation program—Users guide." NBSIR 85-3243. U.S. Department of Commerce, National Bureau of Standards, National Engineering Laboratory, Center for Building Technology, Gaithersburg, Maryland.

Clarke, D. W. (1983). "PID Algorithms and their computer implementation," O.E.U.L. Report N0. 1482/83, Department of Engineering Science, University of Oxford, Oxford, U.K.

Clarke, L., and McLean, D. (1988). "ESP—A building and plant energy simulation system," Version 6, Release 8. Energy Simulation Research Unit, University of Strathclyde, Glasgow, Scotland, U.K.

Cumali, Z. (1988). "Global optimization of HVAC system operations in real time." *ASHRAE Transactions*, v. 94, pt. 1.

Department of Energy/Energy Information Administration (DOE/EIA) (1986a). "Characteristics of commercial buildings 1986." DOE/EIA-0246(86). Superintendent of Documents, U.S. Government Printing Office, Washington DC.

Department of Energy/Energy Information Administration (DOE/EIA) (1986b). "Commercial buildings consumption and expenditures 1986." DOE/EIA-0318(86). Superintendent of Documents, U.S. Government Printing Office, Washington DC.

Department of Energy/Energy Information Administration (DOE/EIA) (1989). "Annual outlook for U.S. electric power 1989." DOE/EIA-0474(89). Superintendent of Documents, U.S. Government Printing Office, Washington DC.

DOE-2 Engineers Manual, Version 2.1A (November, 1982). D.A. York and C.C. Cappiello, eds. Berkeley: Building Energy Simulation Group, Lawrence Berkeley Laboratory, and Solar Energy Group, Energy Division, Los Alamos National Laboratory.

Electric Power Research Institute (EPRI) (1986). "The COMMEND planning system: National and regional data and analysis." Prepared by Georgia Institute of Technology. EPRI EM-4486. Palo Alto, California.

Electric Power Research Institute (EPRI) (1987). "ASD directory," Second edition. Palo Alto, California. Cited by Greenberg et al. (1988).

Englander, S. L. and Norford, L. K. (1988). "Fan energy savings: Analysis of a variable speed drive retrofit." Proceedings, American Council for an Energy-Efficient Economy (ACEEE) Summer Study on Energy Efficiency in Buildings, August, 1988, Pacific Grove California.

Eto, J. H. and de Almeida, A. (1987). "Saving electricity in commercial buildings with adjustable speed drives." Report LBL-22629, Lawrence Berkeley Laboratory, Berkeley, California.

Farkash, L. (1989). Personal communication. Environmental Technologies Inc., Largo, Florida.

Feuermann, D. and Kempton, W. (1987). "ARCHIVE: software for management of field data." PU/CEES Report No. 216. Center for Energy and Environmental Studies, Princeton University, Princeton, New Jersey.

Fish, J. (1989, 1990). Personal communication. Titus Products, Division of Philips Industries Inc., Richardson, TX.

Graham, J. B. (1989). "Air Handling," *Technology Menu for Efficient End-Use of Energy*. Vol. 1. Environmental and Energy Systems Studies, Lund University, Lund, Sweden.

Greenberg, S., Harris, J. P., Akbari, H., and de Almeida, A. (1988). "Technology assessment: Adjustable-speed motors and motor drives (residential and commercial sectors)." LBL-25080. Buildings Energy Data Group, Lawrence Berkeley Laboratory, University of California, Berkeley, California.

Haves, P. (1985). "Type 73: Discrete time proportional plus integral controller" and "Type 66: Rate-limited actuator," Fortran source code for HVACSIM+ components. Unpublished, University of Oxford, Oxford, U.K.

Haves, P. (1987). "The application of simulation to the evaluation of building energy control systems." Proceedings, UKSC Conference on Computer Simulation, September, 1987, Bangor, Wales.

Haves, P. and Trewhella, D. (1988). "Towards an environment for HVAC control system evaluation." Proceedings, USER-1 Building Simulation Conference, Society for Computer Simulation, September 1988, Ostend, Belgium.

Haves, P. and Dexter, A. L. (1989). "Simulation of local loop controls," Proceedings, Building Simulation '89, IBPSA, Vancouver, British Columbia, Canada, June, 1989.

Hill, C. R. (1985). "Simulation of a multizone air handler," *ASHRAE Transactions*, v. 91, pt. 1b.

Holman, J. P. (1986). *Heat transfer*, 6th ed. McGraw-Hill, Inc. New York, New York.

Hsieh, E. (1988). "Calibrated computer models and their role in building operation." Master's Thesis, Department of Mechanical and Aerospace Engineering, Princeton University, Princeton New Jersey.

Int-Hout, D. (1989, 1990). Personal communication. Environmental Technologies Inc., Largo, Florida.

Janssen, J. E. (1989). "Ventilation for acceptable indoor air quality." *ASHRAE Journal*, October, 1989, v. 31, no. 10.

Jones, B. L., and Brown, J. E. (1984). "Electrical variable-speed drives." *IEE Proceedings*, V. 131, Pt. A, No. 7.

Kelly, G. E. (1990). Personal communication. Building Equipment Division, National Institute of Standards and Technology.

Kelly, G. E. and May, W. B. (1990). "The concept of an emulator/tester for building energy management system performance evaluation." Draft, to be presented at the ASHRAE Winter Meeting, Atlanta, GA.

Klein, S. A., Beckman, W. A., and Duffie, J. A. (1976). "*TRNSYS*—a transient simulation program," *ASHRAE Transactions*, v. 82, pt. 2.

Knice, D. (1985). "Ventilation test report, facility: Prudential Energy Project, Princeton, New Jersey." Prepared for Aircon, Inc., Mountainside, New Jersey, by Dynamic Balancing Corp, Paterson, New Jersey.

Letherman, K. M. (1981). *Automatic controls for heating and airconditioning*. International series in heating, ventilation and refrigeration; v.15. Pergamon Press, New York.

Liu, S.T. and Kelly, G.E. (1988). "Knowledge-based input generating program for building system simulation." *ASHRAE Transactions*, v. 94, pt. 1.

Marsh, C. W. (1988). "DDC systems for pressurization, fume hood face velocity and temperature control in variable air volume laboratories." *ASHRAE Transactions*, v. 94, pt. 2.

McQuiston, F. C. and Parker, J. D. (1982). *Heating, ventilating and air conditioning*, 2nd ed. John Wiley and Sons, Inc., New York.

Nesler, C. G., and Stoecker, W. F. (1984). "Selecting the proportional and integral constants in the direct digital control of discharge air temperature." *ASHRAE Transactions*, v. 90, pt. 2b.

Norford, L. K. (1984). "An analysis of energy use in office buildings: the case of Enerplex." PU/CEES Report No. 158. Center for Energy and Environmental Studies, Princeton University, Princeton, New Jersey.

Norford, L. K., Rabl, A., Socolow, R. H. (1986). "Control of supply air temperature and outdoor airflow and its effect on energy use in a variable air volume system." *ASHRAE Transactions,* V. 92, Pt. 2.

Norford, L. K., Rabl, A., Harris, J., and Roturier, J. (1988). "The sum of megabytes equals gigawatts: energy consumption and efficiency of office PC's and related equipment." Proceedings, American Council for an Energy-Efficient Economy (ACEEE) Summer Study on Energy Efficiency in Buildings, August, 1988, Pacific Grove California.

Norford, L. K. (1989) Personal communication. Massachusetts Institute of Technology, Cambridge Massachusetts.

Ogata, K. (1970). *Modern Control Engineering*. Prentice-Hall, Inc., Englewood Cliffs, New Jersey.

Park, C., Clarke, D. R., and Kelly, G. E. (1985). "An overview of HVACSIM+, a dynamic building/HVAC/control systems simulation program," Proc. 1st. Annual Building Energy Simulation Conference., Seattle, Washington.

Parametrics (1985). "Flow economizer instruction manual IM6845B." Orange, Connecticut. (Now ASEA Brown Boveri).

Percivall, G. (1985). "PI control of a single-duct VAV HVAC system." CERL-TM-E-85/05, AD-A159 049, Technical Manuscript E-85/05, U.S. Army Corps of Engineers Construction Engineering Research Laboratory, Champaign, Illinois.

Pinnella, M. J. (1985). "Modeling, tuning and experimental verification of a fan static pressure control system." Master's Thesis, University of Illinois at Urbana-Champaign.

Rickelton, D. and Becker, H. P. (1972). "Variable air volume." *ASHRAE Journal,* September 1972.

Rohrer, C.E. and Stoecker, W.F. (1986). "Z-transforms as an aid to DDC system analysis." *ASHRAE Transactions,* v. 92, pt. 1b.

Ryan, L. (1986). "Documentation of DAS: A data acquisition software package." Working Paper No. 87. Center for Energy and Environmental Studies, Princeton University, Princeton New Jersey.

Solar Energy Laboratory (1989). *TRNSYS News,* October 1989. University of Wisconsin-Madison.

Sowell, E. F., Buhl, W. F., Erdem, A. E., and Winkelmann, F. C. (1986). "Prototype object-based system for HVAC simulation." Proceedings, Second International Conference on System Simulation in Buildings, December, 1986, Liege, Belgium. Cited by Sowell and Buhl (1988) and Haves (1987).

Sowell, E. F., and Buhl, W. F. (1988). "Dynamic Extension of the Simulation Problem Analysis Kernel (SPANK)." Proceedings, Building Simulation '89, IBPSA, Vancouver, British Columbia, Canada, June, 1989.

Stoecker, W. F. and Stoecker, P. A. (1989). *Microcomputer control of thermal and mechanical systems.* Van Nostrand Reinhold, New York.

The Trane Company (1975). "Centrifugal fans, sizes 12-89, single and double width, classes I, II, and III. DS-FAN6-1175. La Crosse, Wisconsin.

The Trane Company (1982). "Fans and their application in air conditioning." ED-FAN-882. La Crosse, Wisconsin.

The Trane Company (1982b). "Building pressurization control." AM-CON-17-882. La Crosse, Wisconsin.

Tsal, R. J. (1986). "Evaluation of duct design methods." *ASHRAE Transactions*, v. 92, pt. 1a.

Tsal, R. J. (1990). Personal communication. NETSAL and Associates, Huntington Beach, California.

Tsal, R. J. and Behls, H. F. (1989). "Ducting," *Technology Menu for Efficient End-Use of Energy*. Vol. 1. Environmental and Energy Systems Studies, Lund University, Lund, Sweden.

Tsal, R. J., Behls, H. H., and Mangel, R. (1988). "T-Method duct design, part 1: Optimization theory." *ASHRAE Transactions*, v. 94, pt. 2.

Walker, C.A. (1984). "Application of direct digital control to a variable air volume system." *ASHRAE Transactions*, v. 90, pt. 2b.

Warren, M. (1989, 1990). Personal communication. ASI Controls, San , California

Watson, J. (1988). Personal Communication. Johnson Controls, Philadelphia, Pennsylvania.

Wise, D. (1989). Personal communication. The Trane Company, La Crosse, Wisconsin.

Wood, W. R. (1987). "Beware of pitfalls when applying variable-frequency drives." *Power*, February 1987.

Woodward, R. (1989). Personal communication. ASEA Brown Boveri, Orange, Connecticut.

Appendix A

Instrumentation and Data Acquisition

The instrumentation used in this study to measure fan performance is part of a whole-building data acquisition system that is described in detail in a previous study of the building (Norford, 1984); calibration of supply and return airflow sensors for this building has been documented by Curtiss (1987). The pertinent features of the hardware are described here, as well as the software developed to perform the data acquisition, and notes on problems with some of the measurements.

The existing system (Figure A.1) consists of data acquisition software (Ryan, 1986) running on a microcomputer (the "front end"), connected by serial link to a receiver/transmitter which communicates with remote boards performing analog to digital conversion and multiplexing of signals from sensors. For measurements of terminal box performance, a terminal box communications network typical of that used in DDC zone control systems was installed and interfaced with the existing system. A separate power line carrier (PLC) system, synchronized with the main system, was installed for the purpose of instrumenting the offices served by the terminal box, and also for additional parameter measurements on the terminal box itself not handled by the box controller. Figure A.1 shows the general layout of the systems used; a list of sensors is given in

Tables 3.1 and 3.2 in Chapter 3. Some of the data listed in Table 3.2 that are terminal box operating parameters are stored on the box controller board. Other parameters are stored there as well; only the ones pertinent to this study, however, are listed here.

The terminal box network, interface, and gateway software are part of a standard package provided by the terminal box manufacturer, Environmental Technologies. This package also includes front end zone management software used for terminal box configuration and zone scheduling, but does not record time histories of measured data; for this purpose the data acquisition software running on the front end computer was modified to communicate with the gateway. Hardware-specific portions of the software developed in the course of this work are included in the three listings at the end of this appendix.

The PLC system works by sending digital signals in high-frequency bursts over a power line, so that dedicated wiring between the sensors and the front end can be eliminated. In this case, modifications to building wiring to permit signals to cross between circuits was not worth the effort required given the short duration of the experiments; therefore the transponders were located in a room with the computer, and shielded cable was run through the ceiling plenum to the test area.

All sensors were calibrated in the laboratory or the field. In two instances, room temperatures and fan electricity (at one-minute intervals), low data resolution was a problem:

Room temperature was measured both by the PLC system and the terminal box. There was good agreement between the two measurements; unfortunately the resolution of each measurement was low, about 1 °F.[1] The rough nature of the resulting data is visible Figures 3.16 and 3.18 of Chapter 3. Although this problem was not anticipated enough in advance of the experiments to be remedied, some amelioration was possible

[1]The range of the sensors used in the PLC system was too wide, and only integer values (°F) were returned by the gateway.

through use of desired airflow rate returned by the terminal box—during periods when room temperature deviated from set point, the desired flow rate, proportional to this deviation and reported with twice as much resolution, could be used to calculate it. Smoothing was tried, but was not effective due to the nature of the narrow range of temperatures observed.

In the case of fan electricity, the low resolution was a result of infrequent pulses from power meters designed to be used over longer intervals. Fortunately, applying binomial smoothing in time series successfully removed most of the aliasing, as shown in the fan power time series data presented in Chapters 2 and 3.

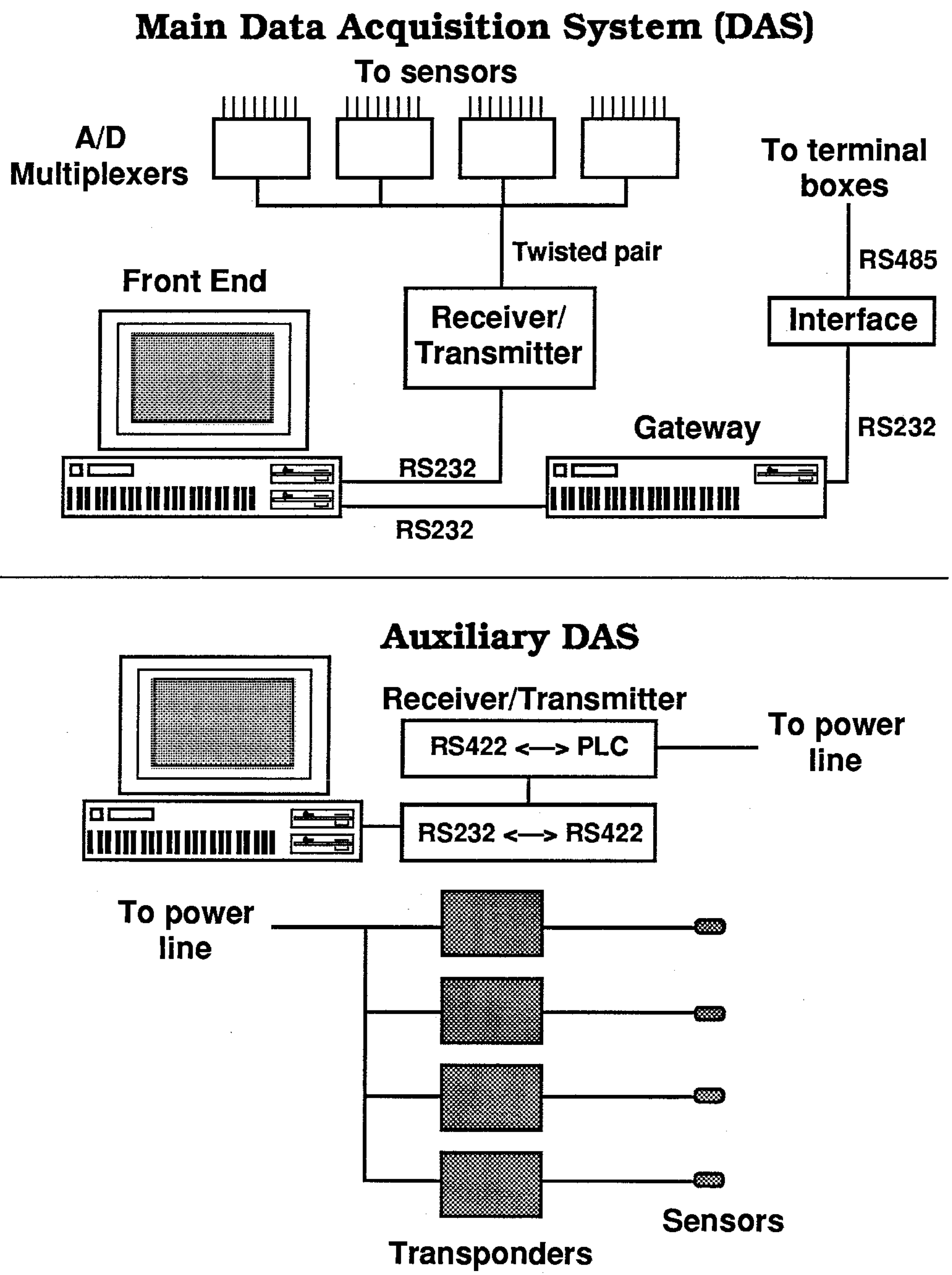


Figure A.1. General configuration of the data acquisition systems used to collect data in this study. The front end computer uses polled communication to exchange data with the main DAS multiplexers and the zone management system gateway. The gateway (another microcomputer) uses interrupt-driven communication to poll the terminal boxes (one in this case), constantly updating the information it has stored for each address. Commands to configure the boxes or get data are sent by the front end to the gateway. A schedule of operating modes determined by the building operator is stored in the gateway, which will command a box to enter a new mode when necessary. The RS485 interface is usually a peripheral board inside the gateway; here, as shown, the interface doubled as a service tool which could be used to connect a laptop computer to the terminal box through the wall thermostat, for balancing purposes.

The power line carrier system was used to instrument the office space, recording temperatures, light status, and door status, and add additional sensors to the terminal box, such as damper angle, inlet and outlet temperatures, and plenum temperature.

LISTING A.1. PASCAL SOURCE CODE FOR POLLING AND PROCESSING PROCEDURES

```
{PROCESS.PAS - Process procedures for data from Intersil REMDACS and ETI
      zone management gateway}
{by Scott Englander, portions by Laurie Ryan, Art McGarity, Peter Curtiss, and
      Environmental Technologies Inc.}
{for Turbo Pascal 5.0}

procedure update_data(i: integer);
begin                         {Increase poll count, update cumulative
reading}
  WITH channel[i] DO
  begin
    std_counter:=std_counter+1;   std_data:=std_data+cur_reading;
    tot_counter:=tot_counter+1;   sqr_std_data:=sqr_std_data+SQR(cur_reading);
    if opt_period <>0 then                    {check for optional data}
    begin
      opt_counter:=opt_counter+1; opt_data:=opt_data+cur_reading;
      sqr_opt_data:=sqr_opt_data+SQR(cur_reading);
    end;
  end;
end;

procedure process_analog(i,data: integer);      {Process analog channel data}
begin
  data:=data shr 4;             {Mask address information - Hardware specific}
  IF data and 2048>1 THEN
    data:= -((data-1) xor 4095);    {2's comp. conversion}
  with channel[i] do
  IF abs(last_data-data) <= maxjump THEN
   begin                       {Check for runtime channel - Hardware specific}
     IF (ch_type = 3) THEN
       IF (data > 2000) OR (data < 0) THEN
         data := 1
       ELSE data := 0;
     cur_reading := slope * data + intercept;
     update_data(i);
   end;
  channel[i].last_data := data;
end;

procedure process_digital(i,data: integer);    {Process digital channel data}
VAR delta: real;
begin
  WITH channel[i] DO
   begin
     IF cur_reading = missing THEN
       delta := 0
     ELSE
       delta := data - cur_reading;          {last polled}
     IF delta<0 THEN
       delta := data+maxcount - cur_reading;      {Counter overflow}
     cur_reading := data;
     IF delta <= maxjump THEN
       begin
         std_counter := std_counter + 1;              {Update polled count}
         tot_counter := tot_counter + 1;
         std_data:= std_data + delta*slope;
         IF opt_period <> 0 THEN                  {check for optional data}
           begin
             opt_counter := opt_counter + 1;
             opt_data := opt_data + delta * slope;
           end;
        end;
    end;
end;

function crc_update(crc : longint; crc_char : byte) : longint;
{
     By Environmental Technologies, Inc.  Converted from C to Pascal by
```

```
  Scott Englander.

     This function must be called once for each character which is to be
  included in the CRC for messages to be transmitted.  This function is
  called once for each character which is included in the CRC of a received
  message, AND once for each of the two CRC characters at the end of the
  received message.  If the resulting CRC is zero, then the message has been
  correctly received.

     x will contain the character to be processed in bits 0-7 and the CRC
  in bits 8-23.  Bit 24 will be used to test for overflow, and then cleared
  to prevent the sign bit of x from being set to 1.  Bits 25-31 are not used
  (x is treated as though it is a 32 bit register.)

     The loop is repeated once for each bit of the character.  The high-order
  bit of the character is shifted into the low-order bit of the CRC, and the
  high order bit of the CRC is shifted into bit 24.  The high order bit of
  the CRC is tested to see if it is a 1; if so, it is exclusive or'd with a 1
  to set it to 0, and the CRC is also xor'd with hex 1021 to produce the
  CCITT-recommended generator of X**16 + X**12 + X**5 + 1.  (To produce the
  CRC generator of X**16 + X**15 + X**2 + 1, change the constant from
  $01102100 to $01800500.  This will xor the CRC with hex 8005 and produce
  the same CRC that IBM uses for their synchronous transmission protocols.)
  Finally, the unneeded bits are anded off and the result is shifted 8 bits
  to the right.

  calling sequence:
    crc := crc_update(crc,next_char);

}
  const
    generator = $01102100;    { CRC-CCITT standard, used by ETI }
    { generator = $01800500   { CRC-16 standard, used by IBM }
  var
    x : longint;
    i : integer;
    temp : real;
  begin
    temp := crc;              {trunc expects a real argument}
    x := (trunc(temp) shl 8) + crc_char;
    for i := 0 to 7 do
      begin
        x := x shl 1;
        if (x and $01000000) > 0 then
          x := x xor generator
      end;
    crc_update :=(x and $00ffff00) shr 8
  end;

function crc_finish(crc : longint) : longint;
{ From ETI }
{    This function must be called once after all the characters in a block
  have been processed for a message which is to be TRANSMITTED.  It returns
  the calculated CRC bytes, which should be transmitted as the two characters
  following the block.  The first of these 2 bytes must be taken from the
  high-order byte of the CRC, and the second must be taken from the low-order
  byte of the CRC.  This routine is NOT called for a message which has been
  RECEIVED.

  calling sequence:
    crc := crc_finish(crc);

        crc_update is called twice, passed a character of hex 00 each time,
     to flush out the last 16 bits from the CRC calculation, and return the
     result as the value of this function.
  }

  begin
    crc_finish := crc_update(crc_update(crc,0),0)
  end;

type
```

```
  term_msg_string = array[0..40] of byte;
var
  term_data : term_msg_string;

procedure add_crc(var msg : term_msg_string);
var crc : longint;
    i, msglen : integer;
begin
  crc := 0;
  msglen := msg[0];
  for i := 1 to msglen do
    crc := crc_update(crc,msg[i]);
  crc := crc_finish(crc);
  msg[msglen + 1] := crc shr 8;
  msg[msglen + 2] := crc and $FF;
  msg[0] := msglen + 2;  {adjust length of msg for crc bytes}
end;

function CRCgood(msg : term_msg_string;
                 var bad_data : boolean) : boolean;
var crc : longint;
    i : integer;
begin
  crc := 0;
  for i := 1 to msg[0] do
    crc := crc_update(crc,msg[i]);
  bad_data := (crc <> 0);
  CRCgood := (crc = 0);
end;

function GroovyData(msg : term_msg_string;
                    var bad_data : boolean;
                    var code : byte) : boolean;
begin
  code := 0;
  if (msg[5] and 32) = 32 then code := 5; {command failure}
  if (msg[5] and 8) = 8 then code := 3;   {gateway busy}
  bad_data := bad_data or (code <> 0);
  GroovyData := (code <> 0)
end;

procedure SendTermMsg(var msg : term_msg_string;
            var timeout : boolean);
var bdummy : boolean;
    i : integer;
    templ,temp2 : boolean;
begin
  for i := 1 to msg[0] do writeport(termport,msg[i]);  {send message}
  for i := 1 to 4 do readport(termport,msg[i],templ);  {read first 4 bytes}
  for i := 5 to 5 + msg[4] do readport(termport,msg[i],temp2);  {get rest }
  msg[0] := i;  {adjust length}
  timeout := templ or temp2
end;

procedure poll_trmbx(i : integer;
                     var msg: term_msg_string;
                     var bad_data : boolean);
  {get data from terminal box}
var
  j: integer;
  code : byte;
begin
  bad_data := false;
  with channel[i] do
  begin
    msg[1] := $AA;          {sync byte}
    msg[2] := 1;            {gateway address}
    msg[3] := number;       {controller address}
    msg[4] := 4;            {number of bytes following}
    msg[5] := 0;            {no data record or description string}
    if ch_type = 40 then
      msg[6] := 1           {request for high priority data}
```

```
    else
      msg[6] := 2;            {request for operational data}
    msg[0] := 6;              {set length of msg string}
  end;
  add_crc(msg);
  SendTermMsg(msg,bad_data);
  if not bad_data then
    begin
      if not CRCgood(msg,bad_data) then
        begin
          error_message(3,'CRC error during communication with gateway');
          code := 8;
        end
      else
        if not GroovyData(msg,bad_data,code) then
          case code of
            5 : error_message(3,'Command failed');
            3 : error_message(3,'Gateway busy')
          end;
      if not bad_data then begin
        for j := 1 to msg[7] do  {byte 7 contains length of data}
          msg[j] := msg[j + 7];  {shift returned data into lower bytes of msg}
        msg[0] := j;             {adjust length byte}
      end
    end
    else
      begin
        error_message(3,'Timeout on communication from gateway');
        code := 9;
      end;
  end;

procedure ProcTermData(i : integer;
                       term_data : term_msg_string);
begin
with channel[i] do begin
  CASE ch_type OF
    {Terminal box channels follow}
    41: {space temp}
        data := term_data[1];
    42: {desired CFM}
        data := term_data[2] + (term_data[3] shl 8);
    43: {actual CFM}
        data := term_data[4] + (term_data[5] shl 8);
    44: {fan speed %}
        data := term_data[6];
    45: {heat stages %}
        data := term_data[7];
    46: {current mode}
        data := term_data[8];
    47: {temperature setpoint}
        data := term_data[9];
    51: {max heating CFM for morning warmup}
        data := term_data[1] + (term_data[2] shl 8);
    52: {min heating CFM for morning warmup}
        data := term_data[3] + (term_data[4] shl 8);
    53: {max cooling CFM for daytime}
        data := term_data[5] + (term_data[6] shl 8);
    54: {min cooling CFM for daytime}
        data := term_data[7] + (term_data[8] shl 8);
    55: {VVF fan start temp offset or airflow}
        data := term_data[9] + (term_data[10] shl 8);
    56: {max heating CFM for daytime}
        data := term_data[11] + (term_data[12] shl 8);
    57: {min heating CFM for daytime}
        data := term_data[13] + (term_data[14] shl 8);
    58: {remote flag for morning warmup}
        data := term_data[15];
    59: {remote flag for daytime}
        data := term_data[16];
    60: {remote flag for night occupied}
        data := term_data[17];
```

```
    61: {remote flag for night unoccupied}
        data := term_data[18];
    62: {warmup temp setpoint (local) or warmup max temp (remote)}
        data := term_data[19];
    63: {undefined (local) or warmup min temp (remote)}
        data := term_data[20];
    64: {day temp setpoint (local) or day max temp (remote)}
        data := term_data[21];
    65: {undefined (local) or day min temp (remote)}
        data := term_data[22];
    66: {night occ. temp setpoint (local) or night occ. max temp (remote)}
        data := term_data[23];
    67: {undefined (local) or night occ. min temp (remote)}
        data := term_data[24];
    68: {night unocc. temp setpoint (local) or night unocc. max temp (remote)}
        data := term_data[25];
    69: {undefined (local) or night unocc. min temp (remote)}
        data := term_data[26];
    70: {Daytime cooling fan control method}
        data := term_data[27];
    71: {Daytime cooling fan speed min}
        data := term_data[28];
    72: {Daytime cooling fan speed max}
        data := term_data[29];
  end; {case}
  IF abs(last_data-data) <= maxjump THEN
    begin
      cur_reading := slope * data + intercept;
      update_data(i)
    end;
  channel[i].last_data := data;
end {with}
end;

procedure special_case(i : integer);
var
  retries : integer;
begin                          {Special channels}
 with channel[i] do
 CASE ch_type OF
  10: {South building heat pump 1 delta T}
      IF (channel[13].cur_reading > 0) AND (channel[13].std_counter > 0) THEN
        begin
          cur_reading:=channel[3].cur_reading - channel[2].cur_reading;
          update_data(i)
        end
      ELSE cur_reading:= missing;
  11: {South building heat pump 2 delta T}
      IF (channel[14].cur_reading > 0) AND (channel[14].std_counter > 0) THEN
        begin
          cur_reading:=channel[8].cur_reading - channel[7].cur_reading;
          update_data(i)
        end
      ELSE cur_reading:= missing;
  30: {Ice pond run time - on considered if flow is greater than 75 gpm }
      IF (channel[114].cur_reading > 75) AND (channel[114].std_counter > 0)
        then begin
          cur_reading:= 1;
          update_data(i)
        end
      ELSE cur_reading:=0;
  40: {get terminal box high priority data}
      begin
        retries := 0;
        repeat
          poll_trmbx(i,term_data,bad_data);
          if not bad_data then
            begin
              cur_reading := 1.0;
              update_data(i);
            end
          else
```

```
            begin
              cur_reading := missing;
              retries := retries + 1;
            end
         until (retries > 3) or not bad_data;
      end;
  41..49: {process terminal box high priority data}
      if not bad_data then
        ProcTermData(i, term_data)
      else cur_reading := missing;
  50: {get terminal box operational data}
      begin
        retries := 0;
        repeat
          poll_trmbx(i,term_data,bad_data);
          if not bad_data then
            begin
              cur_reading := 1.0;
              update_data(i);
            end
          else
            begin
              cur_reading := missing;
              retries := retries + 1;
            end
         until (retries > 3) or not bad_data;
      end;
  51..72: {process terminal box operational data}
      if not bad_data then
        ProcTermData(i,term_data)
      else cur_reading := missing;
  end;
end;

FUNCTION avg_data(chnum: integer): real;        {Average data for a channel}
begin
 CASE channel[chnum].ch_type OF
  1,3,10..19,30..39,40..72:
    IF channel[chnum].std_counter = 0 THEN avg_data:=0 ELSE
    avg_data:= channel[chnum].std_data/channel[chnum].std_counter;
  2,20..29:
     avg_data:=channel[chnum].std_data;
 end;
end;

FUNCTION std_dev(chnum: integer): real; {standard dev. of data for a channel}
begin
  std_dev:=0;
  WITH channel[chnum] DO
  begin
    CASE ch_type OF
      1,4,5: IF std_counter <> 0 THEN
               IF sqr_std_data/std_counter-SQR(std_data/std_counter)>=0 THEN
                 std_dev:=SQRT(sqr_std_data/std_counter-
SQR(std_data/std_counter))
    end;
  end;
end;

procedure process(i: integer);
begin
  IF channel[i].ch_type <=3 THEN poll_channel(i,data,bad_data);
  CASE channel[i].ch_type OF
    1,3: IF NOT bad_data THEN process_analog(i,data);      {Analog, runtime}
    2:   IF NOT bad_data THEN process_digital(i,data);
{Digital}
    ELSE special_case(i);                {Other, user defined}
  end;
end;

procedure InitAdditParams;  {initialize additional system-specific parameters}
begin
```

```
  termport := trunc(par[1]);
  tbaud := trunc(par[2]);  {parameters for ETI gateway}
  tdata := trunc(par[3]);
  tstop := trunc(par[4]);
  tparity := trunc(par[5]);
end;
```

LISTING A.2. SOURCE CODE FOR MAIN DAS POLLING PROCEDURES

```
{POLLING.PAS - Polling procedures for Intersil REMDACS}
{Modified by Scott Englander from original by Laurie Ryan}
{For Turbo Pascal 5.0}

procedure check_tr_flag(VAR data: byte);
var bdummy : boolean;
begin
 REPEAT                                   {Send ascii 'a' TO check tr flag}
  writeport(dataport,97);
  readport(dataport,data,bdummy);
 UNTIL data AND 1 = 1;                   {Check bit 1 FOR transmitter ready}
end;

procedure init_uart(port_no : integer; var timeout : boolean);
const MAXRETRIES = 100;
VAR
  status, trdata: byte;
  retries : integer;
  bdummy : boolean;
begin
  retries := 0;
  querystatus(port_no,status);
  IF status AND 1 > 0 THEN
  REPEAT                                              {Read extraneous data}
    readport(port_no,trdata,bdummy);
    querystatus(port_no,status);
    retries := retries + 1;
  UNTIL (status AND 1 = 0) or (retries > MAXRETRIES);
  timeout := (retries > MAXRETRIES)
end;

procedure error_message(number: integer; msg : generic_string);
  {Flag hardware errors}
var
  trdata : byte;
  j : integer;
begin
 GOTOXY(15,23);
 HighVideo;
 IF scrblankmin <> -1 THEN
 CASE number OF
   1: begin
        WRITELN('Receiver Transmitter error while polling '+ channel[i].name);
        init_uart(dataport,timeout);
        if not timeout then
          FOR j := 1 TO 3 DO
            check_tr_flag(trdata) {Send 3 checks to initialize}
        else
          error_message(0,'Data port timeout                             ')
      end;
   2: begin
        WRITELN('Bad data received from channel '+ channel[i].name);
        init_uart(dataport,timeout);
        if not timeout then
          FOR j := 1 TO 3 DO
            check_tr_flag(trdata) {Send 3 checks to initialize}
        else
          error_message(0,'Data port timeout                               ')
      end;
 else
   writeln(msg)
 end;
 LowVideo;
end;

procedure init_com;
  {initialize data communications}
var
```

```
  trdata : byte;
  j : integer;
begin
  setupcom(dataport,dbaud,ddata,dstop,dparity);   {Set up data port}
  init_uart(dataport,timeout);
  if timeout then
    error_message(0,'Data port timeout                                  ');
  FOR j := 1 TO 3 DO check_tr_flag(trdata);       {Send 3 checks to
initialize}
  setupcom(termport,tbaud,tdata,tstop,tparity);   {Set up terminal box port}
  init_uart(termport,timeout);
  if timeout then
    error_message(0,'Terminal box gateway timeout                       ')
end;

procedure read_data(VAR data: integer;
                    VAR bad_data: boolean);
var bdummy : boolean;
begin
  writeport(dataport,67);
  readport(dataport,ch2,bdummy);         {Read RBR3, ascii 'C'}
  writeport(dataport,70);
  readport(dataport,ch1,bdummy);         {Read RBR2, ascii 'F'}
  data:=ch1 + 256*ch2;                                     {Concatenate data}
  check_tr_flag(ch1);                                        {One last check}
  IF ch1 AND 128 = 128 THEN bad_data:= true ELSE bad_data:= false;
  IF bad_data THEN error_message(2,'');
end;

procedure send_command(ch1,ch2: byte);          {Send a two character command}
begin
  writeport(dataport,ch1);
  writeport(dataport,ch2);
end;

procedure poll_channel(i:integer;
                       VAR data:integer;
                       VAR bad_data:boolean);
VAR trdata: byte;
begin
  send_command(83,channel[i].zone);            {TBR3 command, remote address}
  CASE channel[i].ch_type OF
    1,3: send_command(86,192+channel[i].number);{TBR2 command, analog station}
    2: send_command(86,143+channel[i].number+1);{TBR2 command, digital
                                                 station}
  end;
  check_tr_flag(trdata);                         {Wait for transmitter ready}
  IF trdata AND 64=0 THEN
    read_data(data,bad_data)                     {Check for system error}
  ELSE  error_message(1,'');                        {System error has occured}
end;
```

LISTING A.3. SOURCE CODE FOR DAS SERIAL COMMUNICATIONS

```
{RS232.PAS}
{Serial communications for IBMPC.  Modified by Laurie Ryan, Art McGarity, and
Scott Englander from RMS software}

{for Turbo Pascal 5.0}

Const
  DLL1 = $3F8;           {Divisor Latch Least significant - COM1}
  DLM1 = $3F9;           {Divisor Latch Most significant}
  LCR1 = $3FB;           {Line Control Register}
  MCR1 = $3FC;           {Modem Control Register}
  LSR1 = $3FD;           {Line Status Register, for DR, RTS (data transfer)}
  MSR1 = $3FE;           {Modem status register, for rings, carrier detect}
  DLL2 = $2F8;           {Same addresses for COM2}
  DLM2 = $2F9;
  LCR2 = $2FB;
  MCR2 = $2FC;
  LSR2 = $2FD;
  MSR2 = $2FE;

procedure querystatus(comport: integer;
                      var status: byte);
begin                                          {Look at line status register}
  case comport of
    1: status := port[LSR1];
    2: status := port[LSR2];
  end;
end;

procedure querymodem(comport: integer;
                     var status: byte);
begin                                         {Look at modem status register}
  case comport of
    1: status := port[MSR1];
    2: status := port[MSR2];
  end;
end;

Procedure SetUpCom(comport,baud,data,stop,parity: integer);
var parameter, lcr, dll, dlm: integer;
Begin
  Case comport of
    1: begin  lcr:=lcr1;  dll:=dll1;  dlm:=dlm1;  end;
    2: begin  lcr:=lcr2;  dll:=dll2;  dlm:=dlm2;  end;
  end;   {case port}
  port[LCR]:= 128;                             {Open for access to Baud rate}
  port[DLL]:= 11520 div (baud div 10);            {Set baud rate 1200 to 9600}
  port[DLM]:= 0;                          {High bit only for lower baud rates}
  parameter:=data-5;                                          {Set data bits}
  if stop = 2 then parameter:= parameter + 4;                 {Set stop bits}
  case parity of 1: parameter:=parameter + 24;                  {Even parity}
              -1: parameter:=parameter + 8;                      {Odd parity}
  end;
  port[LCR]:= parameter;                    {Setup and set last bit back to 0}
end;

procedure readport(comport: integer; var data: byte; var timeout : boolean);
  {Read data}
var status: byte;
    start, current : longint;
    flg : integer;
    i: integer;
begin                                      {Assumes called when expecting input}
  timeout := false;
  i := 0;
  repeat
    querystatus(comport,status);
    i := i+1;
```

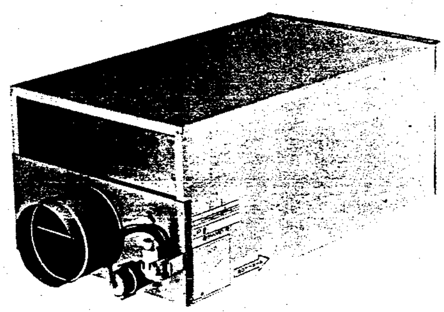
```
  until (status and 1 > 0) or (i>100);
    {Check DR, or timeout }
  if i<100 then
    case comport of
      1: data:=port[DLL1];
      2: data:=port[DLL2];
    end
  else
    begin
      data := 0;
      timeout :=true;
    end;
end;

procedure writeport(comport: integer; data: byte);              {Write data}
var status: byte;
begin
  repeat querystatus(comport,status); until status and 96 = 96;   {Check CTS}
  case comport of
    1: port[DLL1]:=data;
    2: port[DLL2]:=data;
  end;
  repeat
    querystatus(comport,status);
  until status and 96 = 96; {Check til sent}
end;
```

Appendix B
Terminal Box Details

The terminal box used in these experiments was a 14" CVF-II series fan powered terminal with DDC controller, manufactured by Environmental Technologies of Largo, Florida. Included here are catalog excerpts providing technical details of the box, controller, and zone management system.

CONSTANT VOLUME FAN POWERED TERMINALS

MODEL CVF-II



Features

- Low cost, highly efficient waste heat recovery
- Available with electric or hot water auxiliary heat
- Eliminates need for central warm air system
- Single discharge-no need for field fabricated mixing plenum
- Broad range of controls-pneumatic or electronic
- U.L., E.T.L., and C.S.A. listed as an assembly
- Special construction available for low temperature primary air applications (Model CVF-LT-II)

Description

Model CVF-II Fan Induction Terminals are designed for use in low, medium or high pressure variable air volume, single duct systems requiring both cooling and periodic heating of exterior and/or certain interior zones of the building.

The primary air cooling function of the **Model CVF-II** incorporates a single damper blade, which operates through a 45° arc, providing throttling capability in all damper positions-a feature not possible with 90° arc single or multi-blade dampers used in other manufacturers' designs.

The fan induction, or heating function of the **Model CVF-II** provides an inexpensive means of using the waste heat generated in the core of the building by recirculating that energy from the ceiling space to those zones calling for heating. If additional heat is required to maintain zone temperature, the **CVF-II** can be provided with an optional hot water coil or electric heater (**CVF-WC-II**) or (**CVF-EH-II**), which may be activated by the zone thermostat. This eliminates the expense of installing and operating a central warm air heating system. It also allows maximum design flexibility for buildings which experience periodic nighttime or weekend occupancy.

The **Model CVF-II** is a unitary design incorporating both the cooling and heating function in a single casing. Both heated and cooled air pass through a single discharge to downstream ductwork.

Model CVF-II units are available with an optional filter section located at the unit induction port

Construction

Model CVF-II Fan Induction Terminals are manufactured of zinc-coated steel: 22-gauge casing, 20 gauge bottom access door, 16 gauge damper, and 20 gauge damper seat. (Heavier casing gauges are available at extra cost.) Assembly of the casing is by means of a mechanical lock, insuring the tightest possible construction. The damper assembly provides an acoustically effective double wall construction in the high pressure region of the Terminal, which substantially reduces radiated noise at the inlet. Maximum air valve leakage is 2% at 3" w.g.

Units may be provided with round, oval or rectangular inlet collars. Round or oval inlets and rectangular outlets are standard, unless otherwise specified. Convenient bottom access to the terminal interior assembly is provided for component maintenance. Access openings are clearly indicated on dimension drawings. Care should be exercised in maintaining these openings "clear", to insure convenient future access.

Pressure independent units are furnished with an inlet-mounted differential pressure sensor which may be removed without disconnecting the inlet duct.

Model CVF-II casings are internally lined with 3/4" thick, 4# dual density coated fiberglass, complying with N.F.P.A. 90-A and UL 181. No raw edges are exposed to the air stream. Special insulation coatings are available for clean-room, hospital and laboratory applications.

Fan assemblies used in **Model CVF-II** units are specifically designed for fan induction Terminal application. Unlike other manufacturers who use off-the-shelf fan assemblies, ENVIRO-TEC fabricates its fan package using computer selected wheels for specific capacity (CFM) and external static pressure requirements. This insures optimum quiet operation. All fan assemblies are mounted on reinforced casing panels. Fan motors are equipped with spring isolators secured to the fan housing by means of rubber grommets, virtually eliminating vibration transfer.

Electrical components used in the **Model CVF-II** are installed in accordance with UL and N.E.C. requirements. A single-point electrical connection is provided for main power. Standard voltages are 115 or 277, single phase. Special voltages can be provided upon special request.

For thermal storage applications, the construction is altered in three ways: First, the primary air valve is thermally isolated from the fan casing precluding conductance. Secondly, the interior casing insulation is wrapped with a vapor barrier. Finally, the entire bottom casing serves as the access panel. This version is designated as Model CVF-LT-II. Performance is not affected by these alterations. See page 31 for low temperature application information.

Model CVF-II assemblies are UL listed-UL control no. 26H8.
Model CVF-II assemblies are ETL listed-ETL report no. 476203.
Model CVF-II assemblies are CSA listed-CSA file no. LR82026-1.

Performance

Model CVF-II Fan Induction Terminals have been designed with cooling valve and fan assembly matched to provide a constant cooling-to-heating air ratio. If additional heating capacity is required, the **Model CVF-II** can be provided with an integrally mounted hot water

MODEL CVF-II

Performance Cont.

coil or electric heater, which is energized on a call for additional heating through the unit control system.

Model CVF-II units are available as system pressure independent or system pressure dependent. The thermostat controls the **CVF-II** in either case, providing desired temperature by varying the air volume to the space served. Pressure independent models are equipped with minimum/maximum air volume dials for rapid field setting (may also by ordered factory pre-set). Pressure independent models are equipped with a differential pressure inlet averaging sensor to assist in overcoming inlet effect.

When a poor inlet condition exists (other than straight), a shift in the controller set point may occur (if factory set) requiring additional trim adjustment of the controller in the field. With the standard differential controller, flow taps are provided for field setting. System pressure dependent models operate only in response to the room thermostat demand and may fluctuate through their range as the system pressure changes.

System pressure dependent control should be limited to smaller systems where pressures do not vary significantly due to load shedding.

Model CVF-II units will operate efficiently at differential pressures from as low as .03" (Pneumatic) and .015" (Electronic).

Selection

Model CVF-II Fan Induction Terminals should be selected for primary cooling in the mid-range of the performance table (CFM) to insure maximum operating efficiency.

When selecting the proper fan assembly, care should be exercised in determining external static pressure requirements. The fan curves give the external static pressure available at the discharge for each listed size vs. CFM. If an excessively oversized fan assembly is applied, the fan must be throttled to maintain the specified capacity (CFM) at the reduced external Ps requirement, and damaging low fan motor RPM's may result. Conversely, if a smaller than required fan assembly is selected, the unit in all probability will not produce the required external Ps resulting in an under-aired condition; which is expensive to correct in the field.

Various options for fan/motor control are available to meet virtually all requirements. If a unit is properly selected, the standard fan/motor control package will produce the best result. The standard fan motor control package recommended for the **Model CVF-II** includes a 3-tap switch (LOW-MEDIUM-HIGH) in combination with an electronic fan speed controller. This package allows the flexibility of three different horsepower settings and the ability to fine tune fan rpm's for the most efficient operation. In a quick review of the fan performance curves, you can readily see the flexibility provided by the three tap motor selector.

When designing discharge configurations for downstream duct systems, care must be used in the application of the model **CVF-II**. Bullhead tee arrangemets should not be placed less than six feet downstream of the discharge, to allow for proper equalization of air flow and temperature; this will reduce the possibility of stratification. Care should also be exercised in placing diffuser taps too close to the discharge; a similar condition of air shortage can result. It is highly recommended that duct work be designed to provide sufficient pressurization to allow equal flow in the downstream duct system. Splitter dampers in the tee arrangement can cause severe problems where stratification exists. If tee arrangements are employed, linear volume dampers should be used in each leg of the tee and balancing dampers should be provided at each diffuser tap. This arrangement allows maximum flexibility in accomplishing a properly balanced condition.

If you should have any doubt regarding proper discharge configurations, consult your local ENVIRO-TEC Representative, or contact the factory.

Controls

The **Model CVF-II**'s many control sequences represent the broadest range of standard fan powered control options in the industry, providing infinite flexibility to meet any system requirement.

Terminals are available with pneumatic or electronic controls. Control sequence descriptions and reproducible schematics are shown in Control Sequence Guide CSP187-1(Pneumatic), CSE287-1(Electronic) and CSD1088-1 (Direct Digital), located under control section of the general catalog.

Installation

Model CVF-II Fan Induction Terminals are equipped with vibration isolation type motor mountings for maximum reduction of vibration transmission from the casing. Improper installation of the terminal, however, can cause these features to lose their effectiveness.

All **CVF-II** units should be installed in a manner to avoid contact with obstacles such as rigid conduit, sprinkler piping, greenfield, rigid pneumatic tubing, etc.; as such contact can transmit vibration to the building structure, causing objectionable low frequency noise.

Fan terminal units should never be installed tight against concrete slabs or columns, as vibration transmission is amplified in this condition.

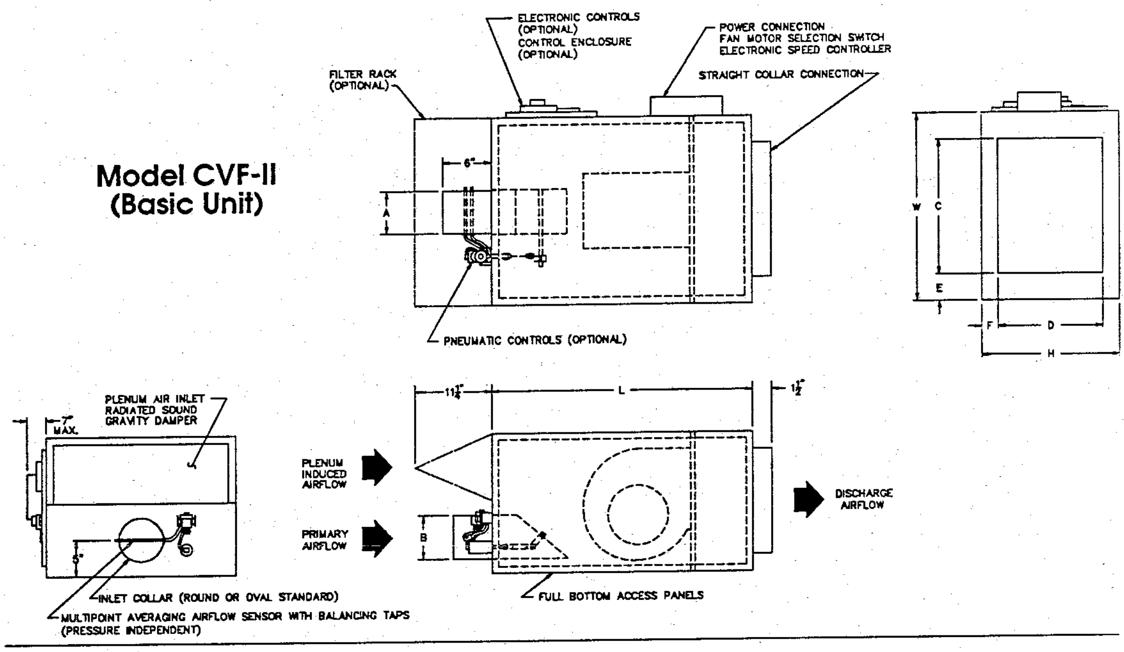
Recommended type hangers: sheetmetal straps securely attached to bar joist or mounting anchors properly secured to slab construction with lugs or poured-in-place hangers. Percussion nails are not considered to be a prudent anchor. Trapeze hangers may also be used, provided rubber liners are used on the contact rails of the hangers, eliminating metal-to-metal contact.

Inlet approaches to system pressure independent **Model CVF-II** units should be as straight as possible to eliminate inlet effect. Averaging probes are provided to offset mild inlet effect. If severe approaches are installed, field trim adjustment of the controller may be required to achieve acceptable air balance of factory pre-set terminals.

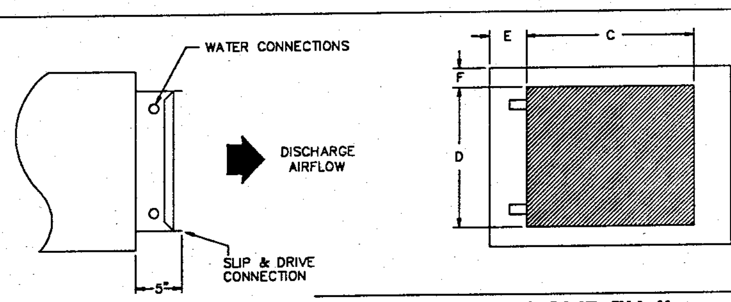
For maximum efficiency in controlling radiated noise in critical applications, we recommend that inlet ducts be fabricated of not less than 24-gauge sheet metal in lieu of flexible duct connections. Flexible duct is extremely transparent to radiated sound; consequently high inlet static pressures (Ps) or sharp bends with excessive pressure drop can cause a radiated noise problem in the space.

Model CVF-II Dimensional Data

Model CVF-II (Basic Unit)

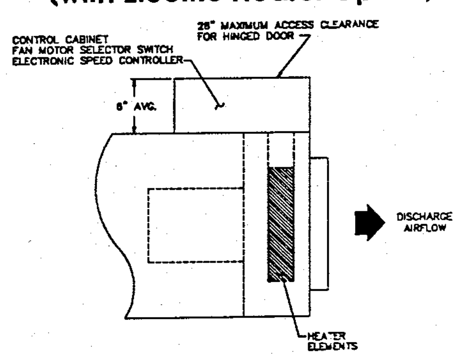


Model CVF-WC-II (with Hot Water Coil Option)



UNIT SIZE	A	B	C	D	E	F	H	L	W	FILTER SIZE	FILTER QTY
6	6"	6"	18"	$11\frac{1}{4}$"	3"	$2\frac{7}{8}$"	17"	40"	24"	12"x24"x1"	2
8	8"	8"	18"	$11\frac{1}{4}$"	3"	$2\frac{7}{8}$"	17"	40"	24"	12"x24"x1"	2
10	11	8"	26"	$11\frac{1}{4}$"	2"	$2\frac{7}{8}$"	17"	40"	30"	12"x30"x1"	2
12	$14\frac{1}{8}$"	8"	26"	$12\frac{1}{2}$"	2"	$3\frac{1}{4}$"	19"	46"	30"	12"x30"x1"	2
14	$17\frac{1}{4}$"	8"	36"	15"	6"	1"	17"	40"	48"	12"x24"x1"	4
16	$20\frac{3}{8}$"	8"	40"	15"	6"	2"	19"	46"	52"	12"x26"x1"	4
18	$23\frac{9}{16}$"	8"	40"	15"	6"	2"	19"	46"	52"	12"x26"x1"	4

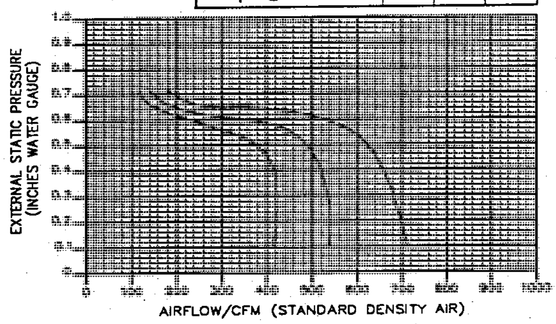
Model CVF-EH-II (with Electric Heater Option)



Model CVF-II Fan Performance Curves

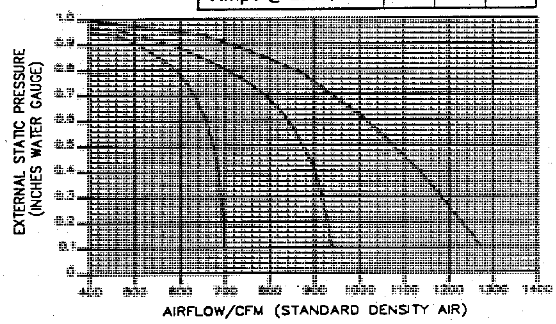
Size 06

Fan Motor Tap	HI	MD	LO
Horsepower (HP)	1/6	1/8	1/10
Amps @ 115 Volts	2.6	1.7	1.3
Amps @ 277 Volts	0.9	0.7	0.5



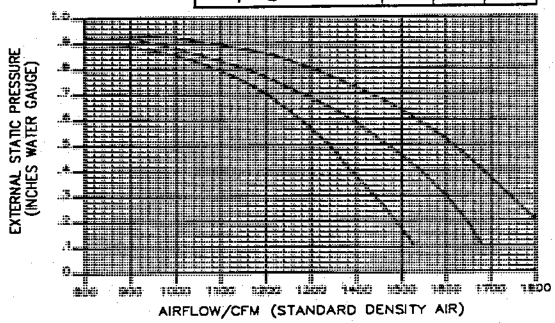
Size 08

Fan Motor Tap	HI	MD	LO
Horsepower (HP)	1/4	1/5	1/8
Amps @ 115 Volts	4.9	2.8	2.0
Amps @ 277 Volts	1.9	1.2	0.8



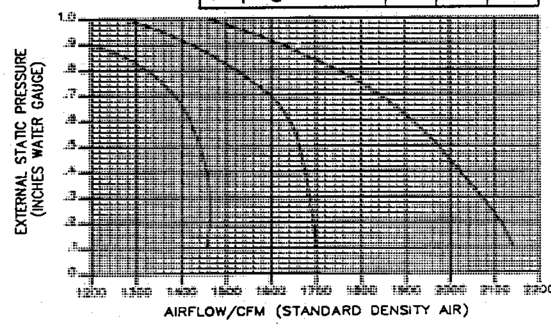
Size 10

Fan Motor Tap	HI	MD	LO
Horsepower (HP)	1/2	1/3	1/4
Amps @ 115 Volts	8.0	6.4	5.0
Amps @ 277 Volts	3.2	2.5	1.9



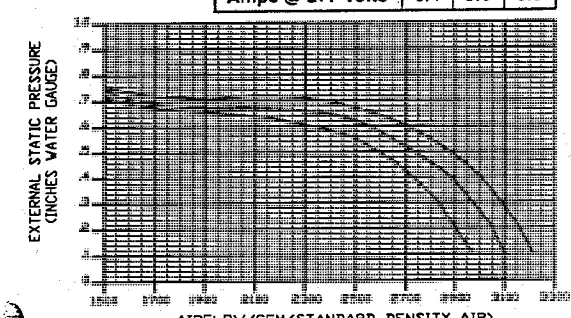
Size 12

Fan Motor Tap	HI	MD	LO
Horsepower (HP)	3/4	1/2	1/3
Amps @ 115 Volts	9.7	7.0	5.4
Amps @ 277 Volts	3.8	2.6	1.9



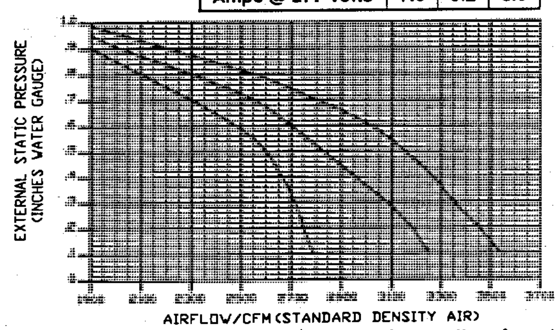
Size 14

Fan Motor Tap	HI	MD	LO
Horsepower (HP)	(2)1/2	(2)1/3	(2)1/4
Amps @ 115 Volts	16.0	12.8	10.0
Amps @ 277 Volts	6.4	5.0	3.8



Size 16

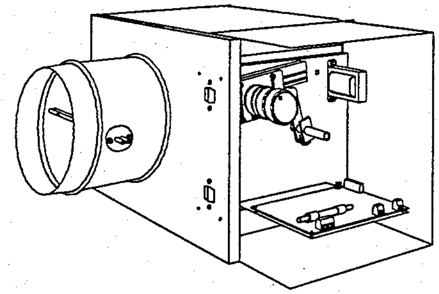
Fan Motor Tap	HI	MD	LO
Horsepower (HP)	(2)3/4	(2)1/2	(2)1/3
Amps @ 115 Volts	19.4	14.0	10.8
Amps @ 277 Volts	7.6	5.2	3.8



Fan curves depict actual performance of each motor tap without any adjustment of the electronic fan speed controller. Actual specified capacities which fall below a certain curve are obtained precisely by adjustment of the electronic fan speed controller.

Unit fans should not be run prior to installation of downstream duct; otherwise, damage to the fan motor may result. The minimum external static pressure requirement is 0.1 inches w.g.

ETDC ZONE CONTROLLER



Function

The Zone Controller is the central processor for a control space. The Zone Controller determines the space temperature requirements and controls room temperature by adjusting the supply damper and optional re-heat devices to satisfy the sensed room load requirements.

Description

The Zone Controller consists of a microprocessor based controller containing Input-Output circuits, communication circuits, the primary supply damper flow sensor, and all necessary power supplies. The program stores configuration parameters in protected memory at the unit, allowing for a safe return from a power failure. The unit is specifically configured for each application, and is capable of being connected directly to Personal Computer Service Tool for real-time display of control processes and controller diagnostics. An onboard LED provides visual operation status information.

Specifications

Type:	Pressure Independent, Electronic, Direct Digital, Microprocessor Controlled
Flow Sensing Method:	Averaging Differential Pressure Sensor and Flow Compensated Differential Pressure Transducer
Range:	0.01 - 1.0 inches of water
Digital Outputs:	24 VAC, 10 VA Max each 2 Standard (Actuator) 4 Optional (Heat or Lighting) 1 Optional (Fan)
Special Optional Outputs:	Stepper Motor Actuator (in lieu of Standard Actuator Outputs above) Variable Fan Speed Control (in lieu of Optional Fan Output above; used with ENVIRO-TEC Direct Digital Fan Speed Controller) Floating, Modulating Hot Water Valve (in lieu of 2 optional Heat Outputs above)

ETDC ZONE CONTROLER

Analog Outputs:	2 Optional, specify from following ranges (contact factory for custom range) 4-20 mADC, 0-5 VDC, 0-10 VDC, 2-10 VDC
Analog Inputs:	Differential Pressure Sensor/Xdcr (standard) Space Sensor (standard) Duct Temperature Sensor (input standard, sensor optional) Up to three additional inputs which may include, but are not limited to: – Temperature Setpoint Adjustment – Duct Temperature Sensor – External Differential Pressure Xdcr – Damper Maximum Position Switch – Pressure Switch
Communications Inputs:	BAS Communications Bus (Standard) Service Tool or Laptop at space sensor (optional) and on controller (standard)
Communications Interface:	RS485; others available on special order
Communications Baud Rate:	9600 baud standard 1200, 2400 or 4800 baud Optional Communications Protocol: Several available for different Building Automation systems; contact factory
Address:	User Selectable by Service Tool or Laptop Computer; Address Switch Optional
Software Features:	Multiple, Stand-Alone control Strategies which may be configured at factory, through Service Tool or Laptop Computer, or through BAS. Active strategy initiated through BAS communications or physical devices (pressure switches, duct sensors, etc.)

Memory Capacity:

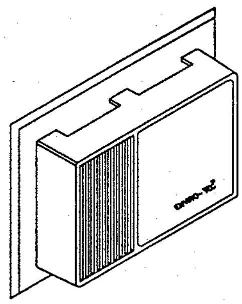
Program (EPROM)	8,192 Bytes Standard 32,768 Bytes Maximum
Random Access (RAM)	2,176 Bytes Standard 8,320 Bytes Maximum
Non-Volatile (EEPROM)	512 Bytes Standard 8,192 Bytes Maximum

Supply Voltage:	24 VAC +10%/-15%, 50/60Hz, 10 VA
Connectors:	Screw Type Terminal Block

Environmental Temperatures:

Storage:	0 to 100 Deg F, 0 to 95% RH, non-condensing
Operating:	35 to 95 Deg F, 10 to 95% RH, non-condensing

ROOM TEMPERATURE SENSOR



Function

The Room Temperature Sensor is located in the Zone, in a location representative of the room temperature (out of direct sunlight, away from discharge air currents). It contains a sensitive, fast-response room temperature sensor, a switch, and an (Optional) communication port. The thermistor provides room temperature data to the temperature control loop of the Zone Controller.

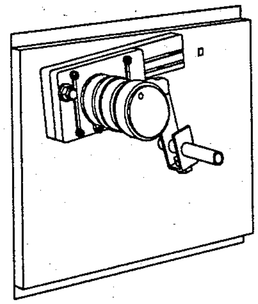
Description

The thermistor is an epoxy encapsulated, hermetically sealed temperature sensor, mounted in a plastic enclosure. A room setpoint adjustment may be provided, (optionally) either under a removable cover, or exposed. A micro-switch is provided to signal after-hours occupancy.

Specifications

Setpoint Range :		50-90 F
Environmental Temperatures:		
	Storage :	-35 to 150 Deg F, 0 to 95% RH, non-condensing
	Operating:	0 to 120 Deg F, 10 to 95% RH, non-condensing
Enclosure :		Plastic
Accuracy:		0.4F @ 77F
Connection:		18 to 20 AWG, stranded copper
Size:	Base:	4.5"w x 3.35"h
	Cover:	3.6"w x 2.5"h, 1.05" deep
Connectors:		Screw Type Terminal Block (Optional communications: 10 pin connector) Number Connectors Required: – STD Temperature & Switch 2 – (Optional) Communication (Shielded Twisted Pair) 1 pair – (Optional) Room Temp 2

STANDARD DAMPER ACTUATOR



Function

The standard electric damper provides actuation for the supply damper.

Description

An AC Synchronous motor connected to a linear pinion gear is connected to the damper shaft. The drive train is a rack and pinion type with lubricated steel gear insert. A pin at the end of the rack attaches to the crank arm on the damper shaft with a Tinnerman clip; crank arm attaches to the damper shaft with a bolt. Full Travel time is Approx. 170 Sec. An (optional) end switch is provided to indicate full open position on the damper for system diagnostics.

Specifications

Supply power:	24 VAC +/- 10% (Provided by Zone Controller) 5 VA
Environmental Temperatures:	
Storage:	-35 to 150 Deg F, 0 to 95% RH, non-condensing
Operating:	0 to 120 Deg F, 10 to 95% RH, non-condensing
Full Travel Time:	170 Sec/45°, Typical
Connections:	Motor wires are factory connected to terminals on the controller board. Yellow is for counter clockwise, blue for clockwise operation. White wires are attached to AC common.
Options:	Damper full open contact closure

COMMUNICATIONS GATEWAY AND ALARM MONITOR (DBM)

Function

The Intel-i-Zone Dedicated Bus Manager (DBM) provides several functions essential to the interface of VAV units and to an Energy Management System.

The functions performed by this unit are broken down as follows:

a. Inter-Unit Communications: The DBM conducts a polling of all the connected units, determining if information needs to be sent to a unit, if a unit needs information, checking status indicators in each unit, and updating the unit time functions. In this regard, it acts as a communications master.

b. Scheduling: The DBM has an on-board battery powered clock, or has access to one through the Building Automation System (BAS). Time schedules are programmed into the DBM, prompting global status messages to be sent to applicable units when a time schedule requires a change in operation status.

c. Interrogation: The DBM provides a "window" into the connected Zone network, providing detailed information on a selected unit. The data may be sent directly to a connected PC "service tool", or to the BAS through the protocol converter.

d. Protocol Conversion: The DBM translates requests for information from the BMS, and acquires the required information. This data is passed back to the BMS in a form suitable to the BMS program. The BMS does not, therefore, communicate directly to an individual controller, although it may appear so to the BMS. Information requests, as well as setpoint modifications from the BMS, are checked for accuracy and validity before passing the data or request on to the unit, providing a check against invalid data being passed to a unit.

e. Alarm Recording: (Optional) The DBM can be equipped with a disk drive to record all alarm messages. The data will be stored in a LOTUS / Spreadsheet format for processing at any time. This data can be of either exceptions (alarms) or status (example: night set-back override times for billing purposes) indicators.

f. TTL Interface: (Optional) the DBM can be equipped to transmit and receive contact closure (TTL, 5V signal) interface informaton with building devices, including BAS systems. These TTL signals can be configured to provide scheduling or override signals to selected zones.

Description

The Communications Gateway is a stand-alone device which requires only a single high voltage power connection and communications connections. It is fully enclosed in a metal housing. If the optional disk drive is supplied, that unit will be easily accessable.

Specifications

Type:	Electronic Direct Digital Microprocessor (8088) Controlled

CONTROL SEQUENCE DESIGNATIONS

Model	Additional Options
ETDSD	Duct Temperature Sensor
ETDSD	Duct Temperature Sensor (2nd sensor)
ETPIOP2	External Flow Transducer Module
ETDO	Damper Open Switch

ETI SERIES 900 SOFTWARE

ENVIRO-TEC has several software products for communicating and controlling the System 900 ETDC controllers. These include the communicating System Monitor program, the Balancing and Calibration Program, and the Factory Configuration and Set-Up program (intended primarily for factory trained personnel).

Direct: One method is to utilize a direct connection to the unit, whether through the optional communications port at the room sensor, or through direct connection at the board. Two programs are available for direct connection:

a. SETCHIP.EXE: ENVIRO-TEC Set-Up and Configuration Program

b. BALANCE.EXE: ENVIRO-TEC Balancing Program

Network: The other method is to access the unit through the ENVIRO-TEC Dedicated Bus Manager (Gateway-DBM), and a personal computer. Two programs are available for network communication through the DBM:

a. FRONT.EXE: ENVIRO-TEC Front/Zone Management System Monitor Program

b. NETBAL.EXE: Network Balancing Program (Available 12/1/88)

Balancing and Calibration Program

Description

The balance program is intended to allow a contract balancer to perform the necessary tasks to insure that a zone controller is measuring the proper quantity of air to the space, and to check (and change if necessary) unit address and to set basic unit performance setpoint data. The program allows for two-point recalibration of the flow sensor output, comparing the unit's displayed flow to on-site measured flow data.

If more detailed configuration changes are required, the specialized program, SETCHIP.EXE, can be used by a factory trained individual to perform low level reset of a unit's configuration parameters.

Requirements

1. Personal Computer ("IBM Compatible") with RS-232 Serial Port, 256K RAM, one floppy diskette drive (either 3.5in 720K or 5.25in 360K).

ETI SERIES 900 SOFTWARE

2. ENVIRO-TEC Service Tool Communications Converter. The converter has a 25 pin serial port connector with 20 ft cord, the battery powered Converter, and a 2 foot cord terminated with the ENVIRO-TEC 10 pin communications plug.
3. BALANCE.EXE program on a appropriate diskette.

Mode:	Daytime
Temperature:	75
Airflow:	200 CFM
Fan:	0
Heat:	0
Box Type:	Single Duct

The Balance program has a number of features. These include a menu of functions as well as a zone monitor:

```
SELECT OPTION

Exit
Read from EEPROM
Change Address
Flow Calibration/Diffuser Balance
Install Setpoints
Write to EEPROM
Hardware Reset
Pause to disconnect from the box
Fan Balance*
```

*If the unit is Fan powered (VVF or CVF), this additional option is provided:

Help screens are provided at the bottom of all screens. The current status block is presented at all times to show current values. The Bottom of the main menu displays the sequence to be followed in using the program:

```
Read data in from the EEPROM. Read box size to confirm that
it is correct. Balance the diffuser. Calibrate the airflow
transducer. Write to EEPROM, then read from EEPROM and con-
firm that the address is correct.
```

A Key Feature of the Balance program is the Calibration Function. This option allows for both calibrating the flow compensated pressure sensor, if necessary, and for balancing the diffusers. The flow calibration procedure consists of setting two desired flow points, one low and one high, and comparing known flow with indicated flow for both values.
A diffuser balance is typically performed at the full volume point satisfying balancing requirements at the same time as the calibration. Diffuser flow measurements

Low Requested:	200 CFM	Low Measured:	210 CFM
High Requested:	800 CFM	High Measured:	815 CFM

```
Enter the actual measured from when the current airflow has
stabalized. Hit <Return> when done.
```

performed with a hood, or using manufacturer supplied area factors are typically used to verify flow values.
In addition to the flow sensor calibration options, a room temperature calibration option is included. All necessary flow setpoints can be input through this program.

System Monitor Program

The ENVIRO-TEC Monitor Program, FRONT.EXE, allows for complete zone management capabilities. Operating from a Personal Computer ("IBM Compatible"), the FRONT program allows for monitoring zone functions, set-up of building schedules in the system Dedicated Bus Managers (DBMs), and alarm processing, all using mouse-compatible "pull-down" menus.

The program functions include the following:

Zone monitoring capability, Zone Screen:

```
                    CALCULATED VARIABLES

Room Temperature:  73             Desired Aiflow:    647 CFM
Actual Airflow:    650 CFM
Percent Heat:      0              Percent Fan:       0
```

```
                      SETPOINTS

Gateway: 1     Addr:     3       Gateway on Line
Group: 1       Acme Industries
               Front Office
Mode:          Daytime
Temperature Setpoint:    73
Box Size:                8
```

The program has multiple "Pull-Down" function menus:

```
Group      Unit       Function        Exit
                      Change Mode
                      Override Temperature
                      Setpoint Adjustment
                      Setup Groups
                      Schedule Setup
                      Upload from Gateway
```

ETI SERIES 900 SOFTWARE

For example, in this case, Change Mode is selected, allowing the operator to change the mode of individual zones by selecting from another menu:

```
SELECT MODE

Smoke Purge
Morning Warm-up
Daytime
Night Unoccupied
Night Occupied
Vacant
```

Group Scheduling

Group Scheduling is based on the ability of the program to assign an individual zone to any number of programmable control groups. These groups can then be assigned to Morning Warm-up, Daytime, and Night Unoccupied modes on a time schedule. Time schedules are established on a weekly time schedule, with a 99 year calendar holiday over-ride.

In addition, groups can be programmed to change mode in response to signal from a contact closure, utilizing the optional TTL interface at one ENVIRO-TEC DBM. Typical programmed responses include Smoke Purge (typically assigned on a per-floor group basis), Morning Warm-up (in response to an optimized start signal from the building Management System, BMS), or Night Occupied Over-Ride from a telephone command system through a BMS.

With prior communications protocol exchange, a BMS can communicate directly with the System DBM units through a serial port interface, utilizing the ENVIRO-TEC scheduling system information with their own software.

Controller Setpoints

The Front program's setpoint option allows limited access to controller setpoints. Full access requires a password security clearance to gain access to another program. The Standard program allows a temporary override of temperature setpoints. Permanent temperature set-point changing requires full access.

With the proper password, the program NETBAL.EXE can be called from the FRONT program. This program has all the BALANCE functions for flow set-point installation, without the calibration option. Using this option, all system control variable settings can be viewed and modified.

Alarming (Available 12/31/88)

The program will allow a number of data points to be trended at set time intervals, and will also allow for recording of alarms based on alarm setpoints established through programmed menus. The output of alarms can be sent directly to a printer, if connected, or can be stored on the floppy diskette in each DBM for downloading later. Data stored on disk is maintained in a format compatible with most spreadsheets for easy preparation of graphical displays of data.

Appendix C
HVACSIM+ Listings

Presented here for reference are component source code listings for the *HVACSIM+* models developed or enhanced in this study: a discrete-time PI controller, used in the terminal box temperature and flow control loops, a rate-limited actuator used to simulate the damper motor, a flow sensor, a speed control, and the heuristic and modified PI fan speed/static pressure minimization controllers. Following these are simulation configuration listings generated by *HVACSIM+* for the simulations presented in Chapters 3 and 4.

LISTING C.1. DISCRETE-TIME PI CONTROLLER WITH DEADBAND[1]

```
C**********************************************************************
C
      SUBROUTINE TYPE72(XIN,OUT,PAR,SAVED,IOSTAT)
C ---------------------------------------------------------------------
C
C     TYPE 72 : FIRST ORDER FLOW SENSOR MODEL
C              by S. Englander, based on existing temp. sensor.
C
C**********************************************************************
      COMMON/CHRONO/ TIME,TSTEP,TTIME,TMIN,ITIME
      COMMON/SOSCOM/ RTOLX,ATOLX,XTOL
      COMMON/XINIT/ INIT,NSAVED
      INTEGER IOSTAT(2)
      DIMENSION XIN(2),OUT(1),PAR(3),SAVED(3)
      WI=XIN(1)
      CWO=XIN(2)
      TCON=PAR(1)
      WZERO=PAR(2)
      WGAIN=PAR(3)
C
      CWI=(WI-WZERO)/WGAIN
      IF(TCON.GE.1.0) GOTO 2000
      IF(ITIME.GT.1) GOTO 3000
      IF(INIT.EQ.0 .OR. SAVED(1).GT.TIME) SAVED(1)=0.
      IF(INIT.EQ.0)SAVED(2)=CWO
3000  IF(TIME.GT.SAVED(1)) SAVED(3)=SAVED(2)
      SAVED(1)=TIME
      CWOUT=CWI
      A=TCON/TSTEP
      IF(A.LT.0.05) GOTO 1000
      DT=CWI-SAVED(3)
      IF(ABS(DT).LT.1.E-10) GOTO 1000
      CWOUT=CWI-DT*EXP(-1./A)
      GOTO 1000
2000  DWODT=(CWI-CWO)/TCON
C
      OUT(1)=DWODT
      IF((IOSTAT(1).LT.-1).OR.(IOSTAT(1).EQ.2)) GOTO 1001
      IOSTAT(1)=0
      RETURN
1001  IOSTAT(1)=0
      IF(ABS(CWI-CWO).LE.RTOLX*ABS(CWO)+ATOLX) IOSTAT(1)=1
      RETURN
1000  OUT(1)=CWOUT
      SAVED(2)=CWOUT
      IF((IOSTAT(1).LT.-1).OR.(IOSTAT(1).EQ.2)) GOTO 1002
      IOSTAT(1)=0
      RETURN
1002  IOSTAT(1)=0
      IF(ABS(CWI-CWOUT).LE.RTOLX*ABS(CWOUT)+ATOLX) IOSTAT(1)=1
      RETURN
      END
```

[1]Based on code by Haves (1985).

LISTING C.2. RATE-LIMITED ACTUATOR, FOR 45° DAMPER[2]

```
C *******************************************************************
C TYPE 67: MODIFIED RATE LIMIT ACTUATOR
C
C   Ph Haves - OU -  3.5.89
C   Modified by S. Englander 891005
C     Modification allows representation of an actuator for which
C     the crank motion is not centered about about the tangent,
C     but begins there.  Also, RANGE is now calculated only once.
C
C *******************************************************************
C * INPUTS                                                          *
C * ------                                                          *
C * 1.  C    :  control signal input to actuator (-)                *
C *                                                                 *
C * OUTPUTS                                                         *
C * -------                                                         *
C * 1.  CV      : actuator position (-)                             *
C * 2.  Y       : valve/damper position (-)                         *
C * 3.  TSSREV  : number of stop/starts/reversals (-)                *
C *                                                      *
C * PARAMETERS                                                      *
C * ----------                                                      *
C * 1.  TTRAN :  travel time (lim-lim) (s)                          *
C * 2.  RESTART: minimum change in demanded position for movement (-) *
C * 3.  HYS :    hysteresis (-)                                     *
C * 4.  CRANG :  crank travel angle (0 for linear)                  *
C *                                                                 *
C *  SAVED                                                          *
C *  -----                                                          *
C *     1) TIME:   time of previous call for rate limit             *
C *     2) CV:     actuator position at previous call               *
C *     3) CV:     actuator position at previous timestep           *
C *     4) C:      control signal at previous call                  *
C *     5) C:      control signal at previous timestep              *
C *     6) SSREV:  +1=forward, -1=backward, 0=stopped at previous call*
C *     7) SSREV:  +1=forward, -1=backward, 0=stopped at prev timestep*
C *     8) TSSREV: total no. of stop/starts/reversals at previous call*
C *     9) TSSREV: total no. of stop/starts/reversals at prev timestep*
C *    10)         hysteresis                                       *
C *    11)         hysteresis                                       *
C *    12)         time of previous call for hysteresis             *
C *    13) RANGE:  range of actuator travel
C *                                                                 *
C *******************************************************************
C
        SUBROUTINE TYPE67(XIN,OUT,PAR,SAVED,IOSTAT)
C
        DIMENSION XIN(1),OUT(3),PAR(4),IOSTAT(3),SAVED(12)
      LOGICAL CON,QUICK
        COMMON/CHRONO/TIME,TSTEP,TTIME,TMIN,ITIME
        DATA DTR /0.01745/
C
```

[2]Based on code by Haves (1985).

```
        C      = XIN(1)
C
        TTRAN  = PAR(1)
        RESTART= PAR(2)
        HYS= PAR(3)
        CRANG  = PAR(4)
C  Limit control signal
        IF(C.LT.0.) C=0.
        IF(C.GT.1.0) C=1.0
C  First time-step
        IF(ITIME.LE.1) THEN
            IF(INIT.EQ.0 .OR. SAVED(1).GT.TIME) SAVED(1)=0.
            IF(INIT.EQ.0) THEN
                SAVED(2)=C
                SAVED(4)=C
                SAVED(6)=0.0
                SAVED(8)=0.0
                SAVED(13)=2.*SIN(DTR*CRANG/2.)
            ENDIF
        ENDIF
        IF(TIME.GT.SAVED(1)) THEN
            SAVED(3)=SAVED(2)
            SAVED(5)=SAVED(4)
            SAVED(7)=SAVED(6)
          SAVED(9)=SAVED(8)
            RANGE=SAVED(13)
        ENDIF
C  Demanded minus actual at previous timestep
         DC=SAVED(5)-SAVED(3)
        IF (ABS(SAVED(7)).LT.0.5.AND.ABS(DC).LT.RESTART) THEN
C    Change in demanded position too small to restart actuator
            CV=SAVED(3)
          IF (ABS(SAVED(7)).GT.0.5) THEN
C      Actuator moved during previous timestep but is now stopped
              SAVED(8)=SAVED(9)+1.0
          ELSE
C      Actuator stopped both now and during previous timestep
            SAVED(8)=SAVED(9)
          ENDIF
        ELSE
C    Actuator still moving or large change demanded - actuator on
            IF(TTRAN.GT.(TSTEP*1.0E-6)) THEN
            QUICK=.FALSE.
                DCMAX=TSTEP/TTRAN
                IF(ABS(DC).GT.DCMAX) THEN
C      Actuator on continuously
                    CV=SAVED(3)+SIGN(DCMAX,DC)
                SAVED(6)=SIGN(1.0,DC)
                IF (ABS(SAVED(6)-SAVED(7)).GT.0.5) THEN
C        Actuator action different from that at previous timestep
                  SAVED(8)=SAVED(9)+1.0
                ELSE
C        Actuator moving continuously both now and during prev timestep
                  SAVED(8)=SAVED(9)
                ENDIF
                ELSE
C      Switch off when deadband reached
                    CV=SAVED(5)
```

```
                 SAVED(6)=0.0
                 IF (ABS(SIGN(1.0,DC)-SAVED(7)).GT.0.5) THEN
C          Actuator action different from that at previous timestep
                   SAVED(8)=SAVED(9)+1.0
                 ELSE
C          Actuator moved continuously in same direction as now during
C          previous timestep
                   SAVED(8)=SAVED(9)
                 ENDIF
                ENDIF
             ELSE
C      Instantaneous response to new control signal
             QUICK=.TRUE.
                 CV=C
             SAVED(6)=1.0
             SAVED(8)=SAVED(9)+1.0
             ENDIF
         ENDIF
         SAVED(1)=TIME
         SAVED(2)=CV
         SAVED(4)=C
C    Non-linearity due to crank, and hysteresis
         IF (CRANG.GT.0.0) THEN
C    Calculate following line at start of simulation
C            RANGE=2.*SIN(DTR*CRANG/2.)
C    Following line is the difference between this and type (1)66
C            CX=0.5+SIN(DTR*CRANG*(CV-0.5))/RANGE
             CX=2.0*SIN(DTR*CRANG*CV/2.0)/RANGE
C    For angle of 45 deg. starting at tangent, use range=90
         ELSE
             CX=CV
         ENDIF
         Y=HYSTER(CX,SAVED(10),SAVED(11),SAVED(12),HYS)
C    Output
       OUT(1)=CV
       OUT(2)=Y
       OUT(3)=SAVED(8)
C    Freezing allowed if position error small and demanded position
C    constant or if actuator responds instantly
         CON=(IOSTAT(1).LT.-1).OR.(IOSTAT(1).EQ.2)
         IF ((ABS(DC).LE.MAX(RESTART,(RTOLX*ABS(CV)+ATOLX))
     ^ .AND.CON).OR.QUICK) THEN
           IOSTAT(1)=1
           IOSTAT(2)=1
           IOSTAT(3)=1
       ELSE
           IOSTAT(1)=0
           IOSTAT(2)=0
           IOSTAT(3)=0
       ENDIF
C
       RETURN
       END
```

LISTING C.3. FLOW SENSOR

```
C*********************************************************************
C
        SUBROUTINE TYPE72(XIN,OUT,PAR,SAVED,IOSTAT)
C
C -------------------------------------------------------------------
C
C       TYPE 72 : FIRST ORDER FLOW SENSOR MODEL
C                 by S. Englander, based on existing temp. sensor.
C
C*********************************************************************
C
        COMMON/CHRONO/ TIME,TSTEP,TTIME,TMIN,ITIME
        COMMON/SOSCOM/ RTOLX,ATOLX,XTOL
        COMMON/XINIT/ INIT,NSAVED
        INTEGER IOSTAT(2)
        DIMENSION XIN(2),OUT(1),PAR(3),SAVED(3)
        WI=XIN(1)
        CWO=XIN(2)
        TCON=PAR(1)
        WZERO=PAR(2)
        WGAIN=PAR(3)
C
        CWI=(WI-WZERO)/WGAIN
        IF(TCON.GE.1.0) GOTO 2000
        IF(ITIME.GT.1) GOTO 3000
        IF(INIT.EQ.0 .OR. SAVED(1).GT.TIME) SAVED(1)=0.
        IF(INIT.EQ.0)SAVED(2)=CWO
3000    IF(TIME.GT.SAVED(1)) SAVED(3)=SAVED(2)
        SAVED(1)=TIME
        CWOUT=CWI
        A=TCON/TSTEP
        IF(A.LT.0.05) GOTO 1000
        DT=CWI-SAVED(3)
        IF(ABS(DT).LT.1.E-10) GOTO 1000
        CWOUT=CWI-DT*EXP(-1./A)
        GOTO 1000
2000    DWODT=(CWI-CWO)/TCON
C
        OUT(1)=DWODT
        IF((IOSTAT(1).LT.-1).OR.(IOSTAT(1).EQ.2)) GOTO 1001
        IOSTAT(1)=0
        RETURN
1001    IOSTAT(1)=0
        IF(ABS(CWI-CWO).LE.RTOLX*ABS(CWO)+ATOLX) IOSTAT(1)=1
        RETURN
1000    OUT(1)=CWOUT
        SAVED(2)=CWOUT
        IF((IOSTAT(1).LT.-1).OR.(IOSTAT(1).EQ.2)) GOTO 1002
        IOSTAT(1)=0
        RETURN
1002    IOSTAT(1)=0
        IF(ABS(CWI-CWOUT).LE.RTOLX*ABS(CWOUT)+ATOLX) IOSTAT(1)=1
        RETURN
        END
```

LISTING C.4. SPEED CONTROL

```
C***********************************************************************
*******
C
      SUBROUTINE TYPE70(XIN,OUT,PAR,SAVED,IOSTAT)
C
C
C
C-----------------------------------------------------------------------
-------
C
C       TYPE 70 : FAN OR PUMP SPEED CONTROL
C             by S. Englander
C             Speed is returned as the product of the control signal
C             (0 to 1) and the parameter, which is the max. speed in
C             RPS.
C
C***********************************************************************
*******
C
         DIMENSION XIN(1),OUT(1),PAR(1),IOSTAT(1)
C
         OUT(1)=XIN(1)*PAR(1)
         IOSTAT(1)=1
C
         RETURN
         END
```

LISTING C.5. HEURISTIC FAN CONTROLLER

```
C***********************************************************************
*
C
      SUBROUTINE TYPE76(XIN,OUT,PAR,SAVED,IOSTAT)
C----------------------------------------------------------------
C
C       Discrete-time heuristic fan controller
C       by S. Englander 891113
C   SAVED(1) = time of previous call
C   SAVED(2) = smoothed output from previous call
C   SAVED(3) = smoothed output from previous timestep
C   SAVED(4) = time of previous controller execution
C   SAVED(5) = unused
C   SAVED(6) = unsmoothed output from previous call
C   SAVED(7) = unsmoothed output from previous sample instant
C   SAVED(8) = error term from previous call
C   SAVED(9) = error term from previous sample instant
C   PAR(1) = increment rate (s^-1)
C   PAR(2) = decrement rate (s^-1)
C   PAR(3) = controller time constant
C   PAR(4) = sample time
C   PAR(5) = lower limit of controller
C   PAR(6) = high limit of controller
C   PAR(7) = initial value of controller output
C   PAR(8) = deadband
C
      DIMENSION XIN(3), OUT(1), PAR(8), SAVED(9), IOSTAT(3)
      REAL*4 LLIMIT, INCRATE, DECRATE
      COMMON/CHRONO/ TIME,TSTEP,TTIME,TMIN,ITIME
      COMMON/SOSCOM/ RTOLX,ATOLX,XTOL
      COMMON/XINIT/ INIT,NSAVED
C
      CTM=XIN(1)
      CTS=XIN(2)
C     Limit inputs:
      IF (CTM.GT.1) CTM=1
      IF (CTM.LT.0) CTM=0
      IF (CTS.GT.1) CTS=1
      IF (CTS.LT.0) CTS=0
      C=XIN(3)
      INCRATE=PAR(1)
      DECRATE=PAR(2)
      TCON=PAR(3)
      TSAMP=PAR(4)
      LLIMIT=PAR(5)
      HLIMIT=PAR(6)
      CINIT=PAR(7)
      DBAND=PAR(8)
C  initialize at beginning of simulation
      IF (ITIME.LE.1) THEN
        IF (INIT.EQ.0 .OR. SAVED(1).GT.TIME) THEN
          SAVED(1)=0.
          SAVED(4)=0.
        ENDIF
        IF (INIT.EQ.0) THEN
```

```
            SAVED(2)=CINIT
            SAVED(6)=CINIT
            SAVED(8)=0.0
          ENDIF
        ENDIF
C  first call of timestep
        IF (TIME.GT.SAVED(1)) THEN
          SAVED(3)=SAVED(2)
        ENDIF
C  run controller if a sample instant
        IF (TIME.EQ.SAVED(4) .OR. TIME.GE.(SAVED(4)+TSAMP)) THEN
C    first call of timestep - update previous sample instant values
          IF (TIME.GT.SAVED(1)) THEN
            SAVED(7)=SAVED(6)
            SAVED(9)=SAVED(8)
          ENDIF
C    controller
          E=CTS-CTM
          IF (E.LE.DBAND) THEN
            E=0.0
          ELSE
            E=E-DBAND
          END IF
          DELTAE=E-SAVED(9)
          IF (E.GT.0.0) THEN
            IF (DELTAE.GE.-DBAND) THEN
              CSS=SAVED(7)+INCRATE*TSAMP
C             WRITE(*,*) 'Incr. fan speed; t=',TIME,'DE=',DELTAE,'
CSS=',CSS
            ELSE
              CSS=SAVED(7)
C             WRITE(*,*) 'Same fan speed; t=',TIME,'DE=',DELTAE,'
CSS=',CSS
            END IF
          ELSE
            CSS=SAVED(7)-DECRATE*TSAMP
C           WRITE(*,*) 'Decr. fan speed; t=',TIME,'DE=',DELTAE,'
CSS=',CSS
          END IF
C         WRITE(*,*) 'Cs=',CTS,' Cm=',CTM,' E=',E,' S9=',SAVED(9),'
S7=',
C    &      SAVED(7)
C  limit CSS to values from LLIMIT to HLIMIT
          IF(CSS.LT.LLIMIT) CSS=LLIMIT
          IF(CSS.GT.HLIMIT) CSS=HLIMIT
          SAVED(4)=TIME
          SAVED(6)=CSS
          SAVED(8)=E
        ELSE
C  not a sample instant, set output to value from previous sample
instant
          CSS=SAVED(6)
          E=0.
        ENDIF
C  limit C to values from LLIMIT to HLIMIT
        IF(C.LT.LLIMIT) C=LLIMIT
        IF(C.GT.HLIMIT) C=HLIMIT
C
```

```
      IF(TCON.GE.1.0) THEN
        DCDT=(CSS-C)/TCON
        OUT1=DCDT
      ELSE
        DC=CSS-SAVED(3)
        OUT1=CSS
        IF((TCON/TSTEP.GE.0.05).AND.(ABS(DC).GE.1.E-10)) THEN
          OUT1=CSS-DC*EXP(-TSTEP/TCON)
        ENDIF
        SAVED(2)=OUT1
      ENDIF
C
      SAVED(1)=TIME
      OUT(1)=OUT1
      IOSTAT(1)=0
      IF(ABS(E).LE.RTOLX*ABS(CTS)+ATOLX) IOSTAT(1)=1
C**MESS WITH FREEZING CONDITIONS AT YOUR OWN RISK.
      RETURN
      END
```

LISTING C.6. DISCRETE TIME MODIFIED PI FAN CONTROLLER

```
C**********************************************************************
*
C
      SUBROUTINE TYPE75(XIN,OUT,PAR,SAVED,IOSTAT)
C-----------------------------------------------------------------
C
C       Discrete-time PI controller w/deadband and constant decay
(experimental)
C       For use as fan speed controller.
C       Modified from type 73 by P. Haves by S. Englander 891011
C         Modifications include addition of deadband and input limiting.
C         Also: ignores negative error.
C       891124: Fixed improper sampling problem
C   SAVED(1) = time of previous call
C   SAVED(2) = smoothed output from previous call
C   SAVED(3) = smoothed output from previous timestep
C   SAVED(4) = time of previous controller execution
C   SAVED(5) = unused
C   SAVED(6) = unsmoothed output from previous call
C   SAVED(7) = unsmoothed output from previous sample instant
C   SAVED(8) = integral term from previous call
C   SAVED(9) = integral term from previous sample instant
C   PAR(1) = proportional gain
C   PAR(2) = integral gain (=proportional gain/integral time)
C   PAR(3) = controller time constant
C   PAR(4) = sample time
C   PAR(5) = lower limit of controller
C   PAR(6) = high limit of controller
C   PAR(7) = initial value of controller output
C   PAR(8) = deadband
C   PAR(9) = decay (+) or growth (-) term (s^-1)
C
      DIMENSION XIN(3), OUT(1), PAR(9), SAVED(9), IOSTAT(3)
      REAL*4 LLIMIT
      COMMON/CHRONO/ TIME,TSTEP,TTIME,TMIN,ITIME
      COMMON/SOSCOM/ RTOLX,ATOLX,XTOL
      COMMON/XINIT/ INIT,NSAVED
C
      CTM=XIN(1)
      CTS=XIN(2)
C     Limit inputs:
      IF (CTM.GT.1) CTM=1
      IF (CTM.LT.0) CTM=0
      IF (CTS.GT.1) CTS=1
      IF (CTS.LT.0) CTS=0
      C=XIN(3)
      GAINP=PAR(1)
      GAINI=PAR(2)
      TCON=PAR(3)
      TSAMP=PAR(4)
      LLIMIT=PAR(5)
      HLIMIT=PAR(6)
      CINIT=PAR(7)
      DBAND=PAR(8)
      DECONST=PAR(9)
```

```
C  initialize at beginning of simulation
         IF (ITIME.LE.1) THEN
           IF (INIT.EQ.0 .OR. SAVED(1).GT.TIME) THEN
             SAVED(1)=0.
             SAVED(4)=0.
           ENDIF
           IF (INIT.EQ.0) THEN
             SAVED(2)=CINIT
             SAVED(6)=CINIT
             SAVED(8)=CINIT
           ENDIF
         ENDIF
C  first call of timestep
         IF (TIME.GT.SAVED(1)) THEN
           SAVED(3)=SAVED(2)
         ENDIF
C  run controller if a sample instant
         IF (TIME.EQ.SAVED(4) .OR. TIME.GE.(SAVED(4)+TSAMP)) THEN
C    first call of timestep - update previous sample instant values
           IF (TIME.GT.SAVED(1)) THEN
             SAVED(7)=SAVED(6)
             SAVED(9)=SAVED(8)
           ENDIF
C    controller
           GAIN=GAINP*(1.+0.5*TSAMP*GAINI)
           IF (GAINI.GT.1.0E-10) THEN
             TINT=GAINP/GAINI
             BETA=(2.0*TINT-TSAMP)/(2.0*TINT+TSAMP)
           ELSE
             BETA=1.0
           ENDIF
           IF (BETA.LE.0.0) THEN
             WRITE(*,1000)
             STOP
           ENDIF
C  Following line changed by S. Englander to include constant decay term
           CINT=BETA*SAVED(9)+(1.0-BETA)*SAVED(7) + DECONST*TSAMP
           E=CTS-CTM
C  Following lines to compute deadband added by S. Englander
C  Modified to ignore negative error
           IF (E.LE.DBAND) THEN
             E=0
           ELSE
             E=E-DBAND
           END IF
C
           CSS=GAIN*E+CINT
C  limit CSS to values from LLIMIT to HLIMIT
           IF(CSS.LT.LLIMIT) CSS=LLIMIT
           IF(CSS.GT.HLIMIT) CSS=HLIMIT
           SAVED(4)=TIME
           SAVED(6)=CSS
           SAVED(8)=CINT
         ELSE
C  not a sample instant, set output to value from previous sample
instant
           CSS=SAVED(6)
           E=0.
```

```
      ENDIF
C  limit C to values from LLIMIT to HLIMIT
      IF(C.LT.LLIMIT) C=LLIMIT
      IF(C.GT.HLIMIT) C=HLIMIT
C
      IF(TCON.GE.1.0) THEN
        DCDT=(CSS-C)/TCON
        OUT1=DCDT
      ELSE
        DC=CSS-SAVED(3)
        OUT1=CSS
        IF((TCON/TSTEP.GE.0.05).AND.(ABS(DC).GE.1.E-10)) THEN
          OUT1=CSS-DC*EXP(-TSTEP/TCON)
        ENDIF
        SAVED(2)=OUT1
      ENDIF
C
      SAVED(1)=TIME
      OUT(1)=OUT1
      IOSTAT(1)=0
C**DISABLE OLD CONDITIONS FOR FREEZING OF CONTROLLER
C** IF(ABS(CSS-C).LE.RTOLX*ABS(C)+ATOLX) IOSTAT(1)=1
C**TRY ALTERNATE CONDITION FOR FREEZING INTEGRAL PORTION
      IF(ABS(E).LE.RTOLX*ABS(CTS)+ATOLX) IOSTAT(1)=1
C**MESS WITH FREEZING CONDITIONS AT YOUR OWN RISK.
      RETURN
1000  FORMAT(' FATAL ERROR IN TYPE 73: INT TIME < 0.5 SAMP TIME')
      END
```

LISTING C.7. SIMULATION CONFIGURATION—DDC TERMINAL BOX AND ZONE

```
HVACGEN - Simulation GENeration Program
Version 1.9B     (02-17-1987)

Terminal box/zone calibration run  - sys4

SUPERBLOCK 1
     BLOCK 1
         UNIT  1      TYPE  5 - DAMPER OR VALVE
         UNIT  2      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
         UNIT  3      TYPE 72 - FLOW SENSOR
         UNIT  4      TYPE 67 - MODIFIED RATE LIMIT ACTUATOR
         UNIT  5      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT  6      TYPE  1 - FAN OR PUMP
         UNIT  7      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
         UNIT  8      TYPE  4 - FLOW MERGE
         UNIT  9      TYPE 74 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
         UNIT 10      TYPE 74 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
         UNIT 11      TYPE  7 - TEMPERATURE SENSOR
         UNIT 12      TYPE  7 - TEMPERATURE SENSOR
         UNIT 13      TYPE 28 - CONSTANT FLOW RESISTANCE
         UNIT 14      TYPE  2 - CONDUIT (DUCT OR PIPE)
     BLOCK 2
         UNIT 15      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT 16      TYPE  6 - FLOW SPLIT
         UNIT 17      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT 18      TYPE 28 - CONSTANT FLOW RESISTANCE
         UNIT 19      TYPE  6 - FLOW SPLIT
         UNIT 20      TYPE 28 - CONSTANT FLOW RESISTANCE

SUPERBLOCK 2
     BLOCK 3
         UNIT 21      TYPE 15 - ROOM MODEL
----------------------------------------------------

UNIT   1       TYPE   5
DAMPER OR VALVE

1      INPUTS:
        FLOW              1 - FLUID MASS FLOW RATE
        PRESSURE          2 - OUTLET PRESSURE
        CONTROL           1 - CONTROL: RELATIVE POSITION OF DAMPER OR
VALVE ( - )

2      OUTPUTS:
        PRESSURE          1 - INLET PRESSURE

3      PARAMETERS:
           0.120000     FLOW RESISTANCE, DAMPER OR VALVE OPEN [1000/(KG
M)]
           0.900000e-01 LEAKAGE PARAMETER (DIMENSIONLESS)
           0.916900     CHARACTERISTIC: 0=>EXP., 1=>LIN.,
INTERMEDIATE=>INTERME
                 0.     MODE: 0=>CLOSED WHEN CONTROL=0; 1=>CLOSED WHEN
CONTROL=
----------------------------------------------------
```

```
UNIT   2       TYPE   3
INLET CONDUIT (DUCT OR PIPE)

1      INPUTS:
        PRESSURE           7 - INLET FLUID PRESSURE
        PRESSURE           1 - OUTLET FLUID PRESSURE
        TEMPERATURE        7 - INLET FLUID TEMPERATURE
        TEMPERATURE        6 - AMBIENT AIR TEMPERATURE
        TEMPERATURE        1 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

2      OUTPUTS:
        TEMPERATURE        1 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        FLOW               1 - FLUID MASS FLOW RATE

3      PARAMETERS:
          0.390000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.411000e-03 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.484000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.185000e-02 VOLUME (M3)
          0.100000e-03 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

UNIT   3       TYPE  72
FLOW SENSOR

1      INPUTS:
        FLOW               1 - INPUT MASS FLOW RATE
        CONTROL            2 - FLOW SENSOR OUTPUT

2      OUTPUTS:
        CONTROL            2 - FLOW SENSOR OUTPUT

3      PARAMETERS:
          0.500000     SENSOR TIME CONSTANT (SEC)
                0.     FLOW OFFSET (kg/s)
           1.56000     FLOW GAIN (kg/s)
 ----------------------------------------------------

UNIT   4       TYPE  67
MODIFIED RATE LIMIT ACTUATOR

1      INPUTS:
        CONTROL            3 - CONTROL SIGNAL INPUT TO ACTUATOR

2      OUTPUTS:
        CONTROL            0 - ACTUATOR POSITION
        CONTROL            1 - VALVE/DAMPER POSITION
        CONTROL            0 - NUMBER OF STOP/STARTS/REVERSALS

3      PARAMETERS:
           180.000     TRAVEL TIME (LIMIT TO LIMIT) (s)
          0.391000e-02 MINIMUM CHANGE IN DEMANDED POSITION (-)
```

```
            0.      HYSTERESIS (-)
         90.0000     CRANK TRAVEL ANGLE (0 FOR LINEAR) (DEGREES)
 ----------------------------------------------------

 UNIT   5      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW                2 - FLUID MASS FLOW RATE
         PRESSURE            4 - OUTLET PRESSURE
         TEMPERATURE         3 - FLUID INLET TEMPERATURE
         TEMPERATURE         6 - AMBIENT TEMPERATURE
         TEMPERATURE         4 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE         4 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE            3 - INLET PRESSURE

 3      PARAMETERS:
        0.520000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
        0.460000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
         18.4000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
        0.400000     VOLUME (M3)
        0.816000e-02 FLOW RESISTANCE [1000/(KG M)]
              0.     HEIGHT OF OUTLET ABOVE INLET (M)
         1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT   6      TYPE   1
 FAN OR PUMP

 1      INPUTS:
         FLOW                2 - MASS FLOW RATE OF FLUID
         PRESSURE            5 - OUTLET PRESSURE
         RVPS                1 - FAN OR PUMP ROTATIONAL SPEED
         TEMPERATURE         4 - INLET FLUID TEMPERATURE

 2      OUTPUTS:
         PRESSURE            4 - INLET PRESSURE
         TEMPERATURE         5 - OUTLET FLUID TEMPERATURE
         POWER               1 - POWER CONSUMPTION

 3      PARAMETERS:
         99.1000     1ST PRESSURE COEFFICIENT
        -45.5000     2ND PRESSURE COEFFICIENT
         8.43000     3RD PRESSURE COEFFICIENT
       -0.522000     4TH PRESSURE COEFFICIENT
              0.     5TH PRESSURE COEFFICIENT
        0.200000     1ST EFFICIENCY COEFFICIENT
              0.     2ND EFFICIENCY COEFFICIENT
              0.     3RD EFFICIENCY COEFFICIENT
              0.     4TH EFFICIENCY COEFFICIENT
              0.     5TH EFFICIENCY COEFFICIENT
        0.241000     DIAMETER (M)
         1.00000     MODE: AIR=1, WATER=2
```

```
----------------------------------------------------

UNIT   7       TYPE   3
INLET CONDUIT (DUCT OR PIPE)

1      INPUTS:
        PRESSURE          6 - INLET FLUID PRESSURE
        PRESSURE          8 - OUTLET FLUID PRESSURE
        TEMPERATURE       6 - INLET FLUID TEMPERATURE
        TEMPERATURE       6 - AMBIENT AIR TEMPERATURE
        TEMPERATURE       8 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

2      OUTPUTS:
        TEMPERATURE       8 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        FLOW              3 - FLUID MASS FLOW RATE

3      PARAMETERS:
                0.      INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                0.      OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                0.      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
                0.      VOLUME (M3)
          0.100000      FLOW RESISTANCE [1000/(KG M)]
                0.      HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
----------------------------------------------------

UNIT   8       TYPE   4
FLOW MERGE

1      INPUTS:
        FLOW              3 - INLET MASS FLOW RATE 1
        FLOW              1 - INLET MASS FLOW RATE 2
        PRESSURE          3 - OUTLET PRESSURE
        TEMPERATURE       8 - INLET TEMPERATURE 1
        TEMPERATURE       1 - INLET TEMPERATURE 2

2      OUTPUTS:
        FLOW              2 - OUTLET MASS FLOW RATE
        PRESSURE          8 - INLET PRESSURE 1
        PRESSURE         10 - INLET PRESSURE 2
        TEMPERATURE       3 - OUTLET TEMPERATURE

3      PARAMETERS:
         0.120000e-03 FLOW RESISTANCE [1000/(KG M)]
----------------------------------------------------

UNIT   9       TYPE  74
DISCRETE-TIME PI CONTROLLER W/DEADBAND

1      INPUTS:
        CONTROL           2 - CONTROLLED VARIABLE
        CONTROL           5 - SET POINT FOR CONTROLLED VARIABLE
        CONTROL           3 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2      OUTPUTS:
```

```
      CONTROL          3 - OUTPUT CONTROL SIGNAL

3     PARAMETERS:
        0.250000      PROPORTIONAL GAIN (DIMENSIONLESS)
        0.900000e-02 INTEGRAL GAIN (SEC-1)
        0.100000      CONTROLLER TIME CONSTANT (SEC)
         10.0000      SAMPLE TIME (SEC)
              0.      LOW LIMIT (DIMENSIONLESS)
         1.00000      HIGH LIMIT (DIMENSIONLESS)
        0.500000      INITIAL OUTPUT (DIMENSIONLESS)
        0.391000e-02 DEADBAND (DIMENSIONLESS)
----------------------------------------------------

UNIT  10      TYPE  74
DISCRETE-TIME PI CONTROLLER W/DEADBAND

1     INPUTS:
      CONTROL          8 - CONTROLLED VARIABLE
      CONTROL          9 - SET POINT FOR CONTROLLED VARIABLE
      CONTROL          5 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2     OUTPUTS:
      CONTROL          5 - OUTPUT CONTROL SIGNAL

3     PARAMETERS:
        -11.3000      PROPORTIONAL GAIN (DIMENSIONLESS)
              0.      INTEGRAL GAIN (SEC-1)
        0.100000      CONTROLLER TIME CONSTANT (SEC)
         10.0000      SAMPLE TIME (SEC)
        0.120000      LOW LIMIT (DIMENSIONLESS)
        0.833000      HIGH LIMIT (DIMENSIONLESS)
              0.      INITIAL OUTPUT (DIMENSIONLESS)
        0.391000e-02 DEADBAND (DIMENSIONLESS)
----------------------------------------------------

UNIT  11      TYPE   7
TEMPERATURE SENSOR

1     INPUTS:
      TEMPERATURE     24 - INPUT TEMPERATURE
      CONTROL          8 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

2     OUTPUTS:
      CONTROL          8 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

3     PARAMETERS:
         60.0000      SENSOR TIME CONSTANT (SEC)
         13.3300      TEMPERATURE OFFSET (C)
         18.8900      TEMPERATURE GAIN (C)
----------------------------------------------------

UNIT  12      TYPE   7
TEMPERATURE SENSOR

1     INPUTS:
      TEMPERATURE     25 - INPUT TEMPERATURE
      CONTROL          9 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)
```

```
 2      OUTPUTS:
         CONTROL            9 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

 3      PARAMETERS:
            0.100000     SENSOR TIME CONSTANT (SEC)
             13.3300     TEMPERATURE OFFSET (C)
             18.8900     TEMPERATURE GAIN (C)
 ----------------------------------------------------

 UNIT  13      TYPE  28
 CONSTANT FLOW RESISTANCE

 1      INPUTS:
         FLOW               1 - FLUID MASS FLOW RATE
         PRESSURE          10 - OUTLET PRESSURE

 2      OUTPUTS:
         PRESSURE           2 - INLET PRESSURE

 3      PARAMETERS:
            0.100000e-03 FLOW RESISTANCE  [1000/(KG M)]
 ----------------------------------------------------

 UNIT  14      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW               2 - FLUID MASS FLOW RATE
         PRESSURE           9 - OUTLET PRESSURE
         TEMPERATURE        5 - FLUID INLET TEMPERATURE
         TEMPERATURE        6 - AMBIENT TEMPERATURE
         TEMPERATURE        9 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE        9 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE           5 - INLET PRESSURE

 3      PARAMETERS:
            0.310000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            0.280000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
             11.1000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
            0.240000     VOLUME (M3)
            0.720000e-01 FLOW RESISTANCE [1000/(KG M)]
                  0.     HEIGHT OF OUTLET ABOVE INLET (M)
             1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  15      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW               2 - FLUID MASS FLOW RATE
         PRESSURE          11 - OUTLET PRESSURE
         TEMPERATURE        9 - FLUID INLET TEMPERATURE
         TEMPERATURE        6 - AMBIENT TEMPERATURE
```

```
        TEMPERATURE     11 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
        TEMPERATURE     11 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        PRESSURE         9 - INLET PRESSURE

 3      PARAMETERS:
          0.120000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.402000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           42.5300     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.489000     VOLUME (M3)
          0.433000e-01 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 -----------------------------------------------------

 UNIT  16       TYPE   6
 FLOW SPLIT

 1      INPUTS:
        FLOW             2 - INLET MASS FLOW RATE
        PRESSURE        15 - OUTLET PRESSURE 1
        PRESSURE        22 - OUTLET PRESSURE 2

 2      OUTPUTS:
        FLOW             4 - OUTLET MASS FLOW RATE 1
        FLOW             7 - OUTLET MASS FLOW RATE 2
        PRESSURE        11 - INLET PRESSURE

 3      PARAMETERS:
          0.605000e-03 FLOW RESISTANCE [1000/(KG M)]
 -----------------------------------------------------

 UNIT  17       TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
        FLOW             4 - FLUID MASS FLOW RATE
        PRESSURE        17 - OUTLET PRESSURE
        TEMPERATURE     11 - FLUID INLET TEMPERATURE
        TEMPERATURE      6 - AMBIENT TEMPERATURE
        TEMPERATURE     17 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
        TEMPERATURE     17 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        PRESSURE        15 - INLET PRESSURE

 3      PARAMETERS:
          0.170000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.180000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           9.93000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.250000     VOLUME (M3)
          0.840000e-02 FLOW RESISTANCE [1000/(KG M)]
```

```
               0.      HEIGHT OF OUTLET ABOVE INLET (M)
          1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ---------------------------------------------------

 UNIT  18       TYPE  28
 CONSTANT FLOW RESISTANCE

 1      INPUTS:
         FLOW                7 - FLUID MASS FLOW RATE
         PRESSURE           23 - OUTLET PRESSURE

 2      OUTPUTS:
         PRESSURE           22 - INLET PRESSURE

 3      PARAMETERS:
           0.308000     FLOW RESISTANCE  [1000/(KG M)]
 ---------------------------------------------------

 UNIT  19       TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW                4 - INLET MASS FLOW RATE
         PRESSURE           18 - OUTLET PRESSURE 1
         PRESSURE           20 - OUTLET PRESSURE 2

 2      OUTPUTS:
         FLOW                5 - OUTLET MASS FLOW RATE 1
         FLOW                6 - OUTLET MASS FLOW RATE 2
         PRESSURE           17 - INLET PRESSURE

 3      PARAMETERS:
           0.450000e-01 FLOW RESISTANCE [1000/(KG M)]
 ---------------------------------------------------

 UNIT  20       TYPE  28
 CONSTANT FLOW RESISTANCE

 1      INPUTS:
         FLOW                5 - FLUID MASS FLOW RATE
         PRESSURE           19 - OUTLET PRESSURE

 2      OUTPUTS:
         PRESSURE           18 - INLET PRESSURE

 3      PARAMETERS:
           0.739000e-01 FLOW RESISTANCE  [1000/(KG M)]
 ---------------------------------------------------

 UNIT  21       TYPE  15
 ROOM MODEL

 1      INPUTS:
         FLOW                5 - MASS FLOW RATE OF VENTILATION AIR
         TEMPERATURE        17 - VENTILATION AIR INLET TEMPERATURE
         TEMPERATURE        26 - TEMP., FULLY MIXED PART OF AIR MASS [SAME
AS OUT(1)]
```

```
        TEMPERATURE     27 - WALL MASS TEMPERATURE (SAME AS SECOND
OUTPUT)
        TEMPERATURE     28 - INTERIOR MASS TEMPERATURE (SAME AS THIRD
OUTPUT)
        TEMPERATURE     24 - SPATIAL AVG. TEMP., PISTON FLOW AIR MASS
(OUTPUT 4)
        POWER            2 - CONDUCTION HEAT FLOW INTO WALL MASS
        POWER            3 - HEAT FLOW DUE TO INTERNAL GAINS

 2      OUTPUTS:
        TEMPERATURE     26 - TEMPERATURE OF FULLY MIXED PORTION OF AIR
MASS
        TEMPERATURE     27 - WALL MASS TEMPERATURE
        TEMPERATURE     28 - INTERIOR MASS TEMPERATURE
        TEMPERATURE     24 - SPATIAL AVG. TEMP., PISTON FLOW PORTION OF
AIR MASS
        TEMPERATURE     29 - AVERAGE ROOM AIR TEMPERATURE
        TEMPERATURE     30 - EXHAUST AIR TEMPERATURE

 3      PARAMETERS:
           48.5000      VOLUME OF ROOM AIR MASS (M3)
           2000.00      THERMAL CAPACITANCE OF WALLS (KJ/C)
           4240.00      THERMAL CAPACITANCE OF INTERIOR MASS (KJ/C)
           1.00000      HEAT TRANSFER COEFF. TIMES AREA FOR WALL MASS
(KW/C)
          0.500000      HEAT TRANSFER COEFF. TIMES AREA FOR INTERIOR MASS
(KW/C
          0.300000      FRACTION OF AIR MASS WHICH IS FULLY MIXED
 ----------------------------------------------------
 Initial Variable Values:

 PRESSURE      1 ->    0.398118    (kPa)
 PRESSURE      2 ->    0.115342e-01(kPa)
 PRESSURE      3 ->          0.    (kPa)
 PRESSURE      4 ->   -0.138213e-01(kPa)
 PRESSURE      5 ->    0.231570    (kPa)
 PRESSURE      6 ->          0.    (kPa)
 PRESSURE      7 ->    0.423500    (kPa)
 PRESSURE      8 ->   -0.352355e-04(kPa)
 PRESSURE      9 ->    0.110566    (kPa)
 PRESSURE     10 ->   -0.426568e-04(kPa)
 PRESSURE     11 ->    0.377929e-01(kPa)
 PRESSURE     12 ->          0.    (kPa)
 PRESSURE     13 ->          0.    (kPa)
 PRESSURE     14 ->          0.    (kPa)
 PRESSURE     15 ->    0.370123e-01(kPa)
 PRESSURE     16 ->          0.    (kPa)
 PRESSURE     17 ->    0.294531e-01(kPa)
 PRESSURE     18 ->    0.705663e-02(kPa)
 PRESSURE     19 ->          0.    (kPa)
 PRESSURE     20 ->          0.    (kPa)
 PRESSURE     21 ->          0.    (kPa)
 PRESSURE     22 ->    0.372479e-01(kPa)
 PRESSURE     23 ->          0.    (kPa)
 FLOW          1 ->    0.560120    (kg/s)
 FLOW          2 ->     1.29638    (kg/s)
 FLOW          3 ->    0.736263    (kg/s)
 FLOW          4 ->    0.948635    (kg/s)
```

```
FLOW             5 ->    0.309013    (kg/s)
FLOW             6 ->    0.639594    (kg/s)
FLOW             7 ->    0.347757    (kg/s)
TEMPERATURE      1 ->     14.0027    (C)
TEMPERATURE      2 ->     18.0000    (C)
TEMPERATURE      3 ->     19.6805    (C)
TEMPERATURE      4 ->     19.6806    (C)
TEMPERATURE      5 ->     19.7487    (C)
TEMPERATURE      6 ->     18.7800    (C)
TEMPERATURE      7 ->     14.4400    (C)
TEMPERATURE      8 ->     24.0000    (C)
TEMPERATURE      9 ->     19.7474    (C)
TEMPERATURE     10 ->     16.0000    (C)
TEMPERATURE     11 ->     19.8014    (C)
TEMPERATURE     12 ->     16.0000    (C)
TEMPERATURE     13 ->     16.0000    (C)
TEMPERATURE     14 ->     16.0000    (C)
TEMPERATURE     15 ->     16.0000    (C)
TEMPERATURE     16 ->     16.0000    (C)
TEMPERATURE     17 ->     18.0200    (C)
TEMPERATURE     18 ->     16.0000    (C)
TEMPERATURE     19 ->     18.0000    (C)
TEMPERATURE     20 ->     18.0000    (C)
TEMPERATURE     21 ->     18.0000    (C)
TEMPERATURE     22 ->     18.0000    (C)
TEMPERATURE     23 ->     18.0000    (C)
TEMPERATURE     24 ->     21.9000    (C)
TEMPERATURE     25 ->     21.1000    (C)
TEMPERATURE     26 ->     22.8000    (C)
TEMPERATURE     27 ->     23.0000    (C)
TEMPERATURE     28 ->     20.0000    (C)
TEMPERATURE     29 ->     21.9000    (C)
TEMPERATURE     30 ->     21.9000    (C)
CONTROL          1 ->    0.263350    (-)
CONTROL          2 ->    0.359052    (-)
CONTROL          3 ->    0.371578    (-)
CONTROL          4 ->    0.248319    (-)
CONTROL          5 ->    0.520928    (-)
CONTROL          6 ->          0.    (-)
CONTROL          7 ->    0.120000    (-)
CONTROL          8 ->    0.457429    (-)
CONTROL          9 ->    0.411329    (-)
RVPS             1 ->     14.8000    (rev/s)
POWER            1 ->    0.353469    (kW)
POWER            2 ->     2.00000    (kW)
POWER            3 ->          0.    (kW)
-----------------------------------------------------
Simulation Error Tolerances:

1       RTOLX=    0.100000e-03       ATOLX=    0.100000e-04
        XTOL=     0.200000e-03       TTIME=    0.500000

SUPERBLOCK 1
2       FREEZE OPTION 0      SCAN OPTION 0

SUPERBLOCK 2
3       FREEZE OPTION 0      SCAN OPTION 0
-----------------------------------------------------
```

```
The following are Boundary Variables in the simulation:

PRESSURE          7
TEMPERATURE       7
TEMPERATURE       6
POWER             2
POWER             3
----------------------------------------------------
The following are the reported variables:


SUPERBLOCK 1        REPORTING INTERVAL    60.0000
PRESSURE          7
FLOW              1
FLOW              5
FLOW              2
TEMPERATURE       9
TEMPERATURE       7
TEMPERATURE       6
CONTROL           1
CONTROL           2
CONTROL           5
CONTROL           3

SUPERBLOCK 2        REPORTING INTERVAL    60.0000
TEMPERATURE      24
POWER             2
POWER             3
```

LISTING C.8. SIMULATION CONFIGURATION—HEURISTIC FAN CONTROLLER

```
HVACGEN - Simulation GENeration Program
Version 1.9B     (02-17-1987)

Fan speed minimization control using heuristic controller - s1208a

SUPERBLOCK 1
     BLOCK 1
         UNIT  1      TYPE  5 - DAMPER OR VALVE
         UNIT  2      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
         UNIT  3      TYPE 72 - FLOW SENSOR
         UNIT  4      TYPE 67 - MODIFIED RATE LIMIT ACTUATOR
         UNIT  5      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT  6      TYPE  1 - FAN OR PUMP
         UNIT  7      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
         UNIT  8      TYPE  4 - FLOW MERGE
         UNIT  9      TYPE 74 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
         UNIT 10      TYPE 74 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
         UNIT 11      TYPE  7 - TEMPERATURE SENSOR
         UNIT 12      TYPE  7 - TEMPERATURE SENSOR
         UNIT 13      TYPE 28 - CONSTANT FLOW RESISTANCE
         UNIT 14      TYPE  2 - CONDUIT (DUCT OR PIPE)
     BLOCK 2
         UNIT 15      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT 16      TYPE  6 - FLOW SPLIT
         UNIT 17      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT 18      TYPE 28 - CONSTANT FLOW RESISTANCE
         UNIT 19      TYPE  6 - FLOW SPLIT
         UNIT 20      TYPE 28 - CONSTANT FLOW RESISTANCE

SUPERBLOCK 2
     BLOCK 3
         UNIT 21      TYPE  1 - FAN OR PUMP
         UNIT 22      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT 23      TYPE  6 - FLOW SPLIT
         UNIT 24      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT 25      TYPE  6 - FLOW SPLIT
         UNIT 26      TYPE  2 - CONDUIT (DUCT OR PIPE)
         UNIT 27      TYPE 19 - FLOW BALANCE CONTROL
         UNIT 28      TYPE 19 - FLOW BALANCE CONTROL
         UNIT 29      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
         UNIT 30      TYPE 19 - FLOW BALANCE CONTROL
         UNIT 31      TYPE 76 - DISCRETE-TIME HEURISTIC FAN CONTROLLER
         UNIT 32      TYPE 70 - FAN OR PUMP SPEED CONTROL
         UNIT 33      TYPE 67 - MODIFIED RATE LIMIT ACTUATOR

SUPERBLOCK 3
     BLOCK 4
         UNIT 34      TYPE 15 - ROOM MODEL
----------------------------------------------------

UNIT    1       TYPE   5
DAMPER OR VALVE

1      INPUTS:
        FLOW              1 - FLUID MASS FLOW RATE
```

```
        PRESSURE          2 - OUTLET PRESSURE
        CONTROL           1 - CONTROL: RELATIVE POSITION OF DAMPER OR
VALVE ( - )

 2     OUTPUTS:
        PRESSURE          1 - INLET PRESSURE

 3     PARAMETERS:
          0.120000     FLOW RESISTANCE, DAMPER OR VALVE OPEN [1000/(KG
M)]
          0.900000e-01 LEAKAGE PARAMETER (DIMENSIONLESS)
          0.916900     CHARACTERISTIC: 0=>EXP., 1=>LIN.,
INTERMEDIATE=>INTERME
                0.     MODE: 0=>CLOSED WHEN CONTROL=0; 1=>CLOSED WHEN
CONTROL=
 ----------------------------------------------------

 UNIT   2      TYPE   3
 INLET CONDUIT (DUCT OR PIPE)

 1     INPUTS:
        PRESSURE          7 - INLET FLUID PRESSURE
        PRESSURE          1 - OUTLET FLUID PRESSURE
        TEMPERATURE       7 - INLET FLUID TEMPERATURE
        TEMPERATURE       6 - AMBIENT AIR TEMPERATURE
        TEMPERATURE       1 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2     OUTPUTS:
        TEMPERATURE       1 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        FLOW              1 - FLUID MASS FLOW RATE

 3     PARAMETERS:
          0.390000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.410000e-03 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.484000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.185000e-02 VOLUME (M3)
          0.100000e-03 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT   3      TYPE  72
 FLOW SENSOR

 1     INPUTS:
        FLOW              1 - INPUT MASS FLOW RATE
        CONTROL           2 - FLOW SENSOR OUTPUT

 2     OUTPUTS:
        CONTROL           2 - FLOW SENSOR OUTPUT

 3     PARAMETERS:
          0.100000     SENSOR TIME CONSTANT (SEC)
                0.     FLOW OFFSET (kg/s)
           1.56000     FLOW GAIN (kg/s)
```

```
-----------------------------------------------------

UNIT    4       TYPE   67
MODIFIED RATE LIMIT ACTUATOR

1       INPUTS:
         CONTROL            3 - CONTROL SIGNAL INPUT TO ACTUATOR

2       OUTPUTS:
         CONTROL            0 - ACTUATOR POSITION
         CONTROL            1 - VALVE/DAMPER POSITION
         CONTROL            0 - NUMBER OF STOP/STARTS/REVERSALS

3       PARAMETERS:
             180.000      TRAVEL TIME (LIMIT TO LIMIT) (s)
            0.391000e-02 MINIMUM CHANGE IN DEMANDED POSITION (-)
                  0.      HYSTERESIS (-)
             90.0000      CRANK TRAVEL ANGLE (0 FOR LINEAR) (DEGREES)
-----------------------------------------------------

UNIT    5       TYPE    2
CONDUIT (DUCT OR PIPE)

1       INPUTS:
         FLOW               2 - FLUID MASS FLOW RATE
         PRESSURE           4 - OUTLET PRESSURE
         TEMPERATURE        3 - FLUID INLET TEMPERATURE
         TEMPERATURE        6 - AMBIENT TEMPERATURE
         TEMPERATURE        4 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

2       OUTPUTS:
         TEMPERATURE        4 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE           3 - INLET PRESSURE

3       PARAMETERS:
            0.520000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            0.460000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
             18.4000      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
            0.400000      VOLUME (M3)
            0.816000e-02 FLOW RESISTANCE [1000/(KG M)]
                  0.      HEIGHT OF OUTLET ABOVE INLET (M)
             1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
-----------------------------------------------------

UNIT    6       TYPE    1
FAN OR PUMP

1       INPUTS:
         FLOW               2 - MASS FLOW RATE OF FLUID
         PRESSURE           5 - OUTLET PRESSURE
         RVPS               1 - FAN OR PUMP ROTATIONAL SPEED
         TEMPERATURE        4 - INLET FLUID TEMPERATURE

2       OUTPUTS:
         PRESSURE           4 - INLET PRESSURE
```

```
        TEMPERATURE        5 - OUTLET FLUID TEMPERATURE
        POWER              1 - POWER CONSUMPTION

 3      PARAMETERS:
            99.1000      1ST PRESSURE COEFFICIENT
           -45.5000      2ND PRESSURE COEFFICIENT
            8.43000      3RD PRESSURE COEFFICIENT
          -0.522000      4TH PRESSURE COEFFICIENT
                 0.      5TH PRESSURE COEFFICIENT
           0.200000      1ST EFFICIENCY COEFFICIENT
                 0.      2ND EFFICIENCY COEFFICIENT
                 0.      3RD EFFICIENCY COEFFICIENT
                 0.      4TH EFFICIENCY COEFFICIENT
                 0.      5TH EFFICIENCY COEFFICIENT
           0.241000      DIAMETER (M)
            1.00000      MODE: AIR=1, WATER=2
 ----------------------------------------------------

 UNIT   7       TYPE   3
 INLET CONDUIT (DUCT OR PIPE)

 1      INPUTS:
        PRESSURE           6 - INLET FLUID PRESSURE
        PRESSURE           8 - OUTLET FLUID PRESSURE
        TEMPERATURE        6 - INLET FLUID TEMPERATURE
        TEMPERATURE        6 - AMBIENT AIR TEMPERATURE
        TEMPERATURE        8 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
        TEMPERATURE        8 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        FLOW               3 - FLUID MASS FLOW RATE

 3      PARAMETERS:
                 0.      INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                 0.      OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                 0.      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
                 0.      VOLUME (M3)
           0.100000      FLOW RESISTANCE [1000/(KG M)]
                 0.      HEIGHT OF OUTLET ABOVE INLET (M)
            1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT   8       TYPE   4
 FLOW MERGE

 1      INPUTS:
        FLOW               3 - INLET MASS FLOW RATE 1
        FLOW               1 - INLET MASS FLOW RATE 2
        PRESSURE           3 - OUTLET PRESSURE
        TEMPERATURE        8 - INLET TEMPERATURE 1
        TEMPERATURE        1 - INLET TEMPERATURE 2

 2      OUTPUTS:
        FLOW               2 - OUTLET MASS FLOW RATE
        PRESSURE           8 - INLET PRESSURE 1
```

```
      PRESSURE         10 - INLET PRESSURE 2
      TEMPERATURE       3 - OUTLET TEMPERATURE

3     PARAMETERS:
         0.120000e-03 FLOW RESISTANCE [1000/(KG M)]
-----------------------------------------------------

UNIT   9      TYPE  74
DISCRETE-TIME PI CONTROLLER W/DEADBAND

1     INPUTS:
      CONTROL           2 - CONTROLLED VARIABLE
      CONTROL           5 - SET POINT FOR CONTROLLED VARIABLE
      CONTROL           3 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2     OUTPUTS:
      CONTROL           3 - OUTPUT CONTROL SIGNAL

3     PARAMETERS:
         0.300000     PROPORTIONAL GAIN (DIMENSIONLESS)
         0.920000e-02 INTEGRAL GAIN (SEC-1)
         0.100000     CONTROLLER TIME CONSTANT (SEC)
          10.0000     SAMPLE TIME (SEC)
               0.     LOW LIMIT (DIMENSIONLESS)
          1.00000     HIGH LIMIT (DIMENSIONLESS)
         0.500000e-01 INITIAL OUTPUT (DIMENSIONLESS)
         0.391000e-02 DEADBAND (DIMENSIONLESS)
-----------------------------------------------------

UNIT  10      TYPE  74
DISCRETE-TIME PI CONTROLLER W/DEADBAND

1     INPUTS:
      CONTROL           8 - CONTROLLED VARIABLE
      CONTROL           9 - SET POINT FOR CONTROLLED VARIABLE
      CONTROL           5 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2     OUTPUTS:
      CONTROL           5 - OUTPUT CONTROL SIGNAL

3     PARAMETERS:
         -11.3000     PROPORTIONAL GAIN (DIMENSIONLESS)
               0.     INTEGRAL GAIN (SEC-1)
         0.100000     CONTROLLER TIME CONSTANT (SEC)
          10.0000     SAMPLE TIME (SEC)
         0.120000     LOW LIMIT (DIMENSIONLESS)
         0.833000     HIGH LIMIT (DIMENSIONLESS)
               0.     INITIAL OUTPUT (DIMENSIONLESS)
         0.391000e-02 DEADBAND (DIMENSIONLESS)
-----------------------------------------------------

UNIT  11      TYPE   7
TEMPERATURE SENSOR

1     INPUTS:
      TEMPERATURE      24 - INPUT TEMPERATURE
      CONTROL           8 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)
```

```
 2      OUTPUTS:
         CONTROL            8 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

 3      PARAMETERS:
            0.100000     SENSOR TIME CONSTANT (SEC)
             13.3300     TEMPERATURE OFFSET (C)
             18.8900     TEMPERATURE GAIN (C)
 ----------------------------------------------------

 UNIT  12      TYPE   7
 TEMPERATURE SENSOR

 1      INPUTS:
         TEMPERATURE       25 - INPUT TEMPERATURE
         CONTROL            9 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

 2      OUTPUTS:
         CONTROL            9 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

 3      PARAMETERS:
            0.100000     SENSOR TIME CONSTANT (SEC)
             13.3300     TEMPERATURE OFFSET (C)
             18.8900     TEMPERATURE GAIN (C)
 ----------------------------------------------------

 UNIT  13      TYPE  28
 CONSTANT FLOW RESISTANCE

 1      INPUTS:
         FLOW               1 - FLUID MASS FLOW RATE
         PRESSURE          10 - OUTLET PRESSURE

 2      OUTPUTS:
         PRESSURE           2 - INLET PRESSURE

 3      PARAMETERS:
            0.100000e-03 FLOW RESISTANCE  [1000/(KG M)]
 ----------------------------------------------------

 UNIT  14      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW               2 - FLUID MASS FLOW RATE
         PRESSURE           9 - OUTLET PRESSURE
         TEMPERATURE        5 - FLUID INLET TEMPERATURE
         TEMPERATURE        6 - AMBIENT TEMPERATURE
         TEMPERATURE        9 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE        9 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE           5 - INLET PRESSURE

 3      PARAMETERS:
            0.310000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            0.280000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
```

```
          11.1000      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
         0.240000      VOLUME (M3)
         0.720000e-01 FLOW RESISTANCE [1000/(KG M)]
               0.      HEIGHT OF OUTLET ABOVE INLET (M)
          1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  15       TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW                2 - FLUID MASS FLOW RATE
         PRESSURE           11 - OUTLET PRESSURE
         TEMPERATURE         9 - FLUID INLET TEMPERATURE
         TEMPERATURE         6 - AMBIENT TEMPERATURE
         TEMPERATURE        11 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE        11 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE            9 - INLET PRESSURE

 3      PARAMETERS:
         0.120000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
         0.402000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          42.5300      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
         0.489000      VOLUME (M3)
         0.433000e-01 FLOW RESISTANCE [1000/(KG M)]
               0.      HEIGHT OF OUTLET ABOVE INLET (M)
          1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  16       TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW                2 - INLET MASS FLOW RATE
         PRESSURE           15 - OUTLET PRESSURE 1
         PRESSURE           22 - OUTLET PRESSURE 2

 2      OUTPUTS:
         FLOW                4 - OUTLET MASS FLOW RATE 1
         FLOW                7 - OUTLET MASS FLOW RATE 2
         PRESSURE           11 - INLET PRESSURE

 3      PARAMETERS:
         0.605000e-03 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  17       TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW                4 - FLUID MASS FLOW RATE
         PRESSURE           17 - OUTLET PRESSURE
```

```
        TEMPERATURE       11 - FLUID INLET TEMPERATURE
        TEMPERATURE        6 - AMBIENT TEMPERATURE
        TEMPERATURE       17 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
        TEMPERATURE       17 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        PRESSURE          15 - INLET PRESSURE

 3      PARAMETERS:
          0.170000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.180000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           9.93000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.250000     VOLUME (M3)
          0.840000e-02 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  18      TYPE  28
 CONSTANT FLOW RESISTANCE

 1      INPUTS:
         FLOW               7 - FLUID MASS FLOW RATE
         PRESSURE          23 - OUTLET PRESSURE

 2      OUTPUTS:
         PRESSURE          22 - INLET PRESSURE

 3      PARAMETERS:
           0.308000     FLOW RESISTANCE  [1000/(KG M)]
 ----------------------------------------------------

 UNIT  19      TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW               4 - INLET MASS FLOW RATE
         PRESSURE          18 - OUTLET PRESSURE 1
         PRESSURE          20 - OUTLET PRESSURE 2

 2      OUTPUTS:
         FLOW               5 - OUTLET MASS FLOW RATE 1
         FLOW               6 - OUTLET MASS FLOW RATE 2
         PRESSURE          17 - INLET PRESSURE

 3      PARAMETERS:
           0.450000e-01 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  20      TYPE  28
 CONSTANT FLOW RESISTANCE

 1      INPUTS:
         FLOW               5 - FLUID MASS FLOW RATE
         PRESSURE          19 - OUTLET PRESSURE
```

```
2      OUTPUTS:
        PRESSURE         18 - INLET PRESSURE

3      PARAMETERS:
          0.739000e-01 FLOW RESISTANCE   [1000/(KG M)]
----------------------------------------------------

UNIT  21      TYPE   1
FAN OR PUMP

1      INPUTS:
        FLOW               8 - MASS FLOW RATE OF FLUID
        PRESSURE          36 - OUTLET PRESSURE
        RVPS               2 - FAN OR PUMP ROTATIONAL SPEED
        TEMPERATURE       14 - INLET FLUID TEMPERATURE

2      OUTPUTS:
        PRESSURE          14 - INLET PRESSURE
        TEMPERATURE       36 - OUTLET FLUID TEMPERATURE
        POWER              4 - POWER CONSUMPTION

3      PARAMETERS:
            1.35590     1ST PRESSURE COEFFICIENT
           0.254500     2ND PRESSURE COEFFICIENT
           -2.27000     3RD PRESSURE COEFFICIENT
           -40.0060     4TH PRESSURE COEFFICIENT
                 0.     5TH PRESSURE COEFFICIENT
                 0.     1ST EFFICIENCY COEFFICIENT
            10.1500     2ND EFFICIENCY COEFFICIENT
           -31.9800     3RD EFFICIENCY COEFFICIENT
                 0.     4TH EFFICIENCY COEFFICIENT
                 0.     5TH EFFICIENCY COEFFICIENT
            2.11000     DIAMETER (M)
            1.00000     MODE: AIR=1, WATER=2
----------------------------------------------------

UNIT  22      TYPE   2
CONDUIT (DUCT OR PIPE)

1      INPUTS:
        FLOW               8 - FLUID MASS FLOW RATE
        PRESSURE          12 - OUTLET PRESSURE
        TEMPERATURE       36 - FLUID INLET TEMPERATURE
        TEMPERATURE       37 - AMBIENT TEMPERATURE
        TEMPERATURE       12 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

2      OUTPUTS:
        TEMPERATURE       12 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        PRESSURE          36 - INLET PRESSURE

3      PARAMETERS:
          0.500000     INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.218000     OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           1402.00     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
           49.6000     VOLUME (M3)
```

```
          0.100000e-05 FLOW RESISTANCE [1000/(KG M)]
           5.20000     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  23      TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW              8 - INLET MASS FLOW RATE
         PRESSURE         31 - OUTLET PRESSURE 1
         PRESSURE         13 - OUTLET PRESSURE 2

 2      OUTPUTS:
         FLOW              9 - OUTLET MASS FLOW RATE 1
         FLOW             12 - OUTLET MASS FLOW RATE 2
         PRESSURE         12 - INLET PRESSURE

 3      PARAMETERS:
          0.100000e-05 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  24      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW              9 - FLUID MASS FLOW RATE
         PRESSURE         32 - OUTLET PRESSURE
         TEMPERATURE      12 - FLUID INLET TEMPERATURE
         TEMPERATURE      37 - AMBIENT TEMPERATURE
         TEMPERATURE      32 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE      32 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE         31 - INLET PRESSURE

 3      PARAMETERS:
          0.120000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.183000     OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           212.000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
           10.0000     VOLUME (M3)
          0.700000e-02 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  25      TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW              9 - INLET MASS FLOW RATE
         PRESSURE         33 - OUTLET PRESSURE 1
         PRESSURE         35 - OUTLET PRESSURE 2
```

```
 2      OUTPUTS:
         FLOW               10 - OUTLET MASS FLOW RATE 1
         FLOW               11 - OUTLET MASS FLOW RATE 2
         PRESSURE           32 - INLET PRESSURE

 3      PARAMETERS:
           0.500000e-01 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  26       TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW               10 - FLUID MASS FLOW RATE
         PRESSURE            7 - OUTLET PRESSURE
         TEMPERATURE        32 - FLUID INLET TEMPERATURE
         TEMPERATURE         6 - AMBIENT TEMPERATURE
         TEMPERATURE         0 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE         0 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE           33 - INLET PRESSURE

 3      PARAMETERS:
             1.14000      INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           0.300000e-01 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
             50.0000      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
             1.40000      VOLUME (M3)
           0.400000e-01 FLOW RESISTANCE [1000/(KG M)]
                  0.      HEIGHT OF OUTLET ABOVE INLET (M)
             1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  27       TYPE  19
 FLOW BALANCE CONTROL

 1      INPUTS:
         FLOW               12 - MASS FLOW RATE DETERMINED IN THIS
SUPERBLOCK
         PRESSURE           38 - PRESSURE WHERE FLOW ENTERS THIS SUPERBLOCK
         FLOW               14 - MASS FLOW RATE DETERMINED IN ANOTHER
SUPERBLOCK

 2      OUTPUTS:
         PRESSURE           13 - PRESSURE WHERE FLOW EXITS THIS SUPERBLOCK

 3      PARAMETERS:
           0.130000e-02 INITIAL FLOW RESISTANCE PARAM. AT TIME = 0
[1000/(KG M)
 ----------------------------------------------------

 UNIT  28       TYPE  19
 FLOW BALANCE CONTROL

 1      INPUTS:
```

```
        FLOW            11 - MASS FLOW RATE DETERMINED IN THIS
SUPERBLOCK
        PRESSURE        38 - PRESSURE WHERE FLOW ENTERS THIS SUPERBLOCK
        FLOW            13 - MASS FLOW RATE DETERMINED IN ANOTHER
SUPERBLOCK

 2      OUTPUTS:
        PRESSURE        35 - PRESSURE WHERE FLOW EXITS THIS SUPERBLOCK

 3      PARAMETERS:
          0.240000     INITIAL FLOW RESISTANCE PARAM. AT TIME = 0
[1000/(KG M)
 ----------------------------------------------------

 UNIT  29       TYPE   3
 INLET CONDUIT  (DUCT OR PIPE)

 1      INPUTS:
        PRESSURE        38 - INLET FLUID PRESSURE
        PRESSURE        14 - OUTLET FLUID PRESSURE
        TEMPERATURE     38 - INLET FLUID TEMPERATURE
        TEMPERATURE     37 - AMBIENT AIR TEMPERATURE
        TEMPERATURE     14 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
        TEMPERATURE     14 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        FLOW             8 - FLUID MASS FLOW RATE

 3      PARAMETERS:
          0.100000e-04 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                0.     OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.100000e-03 THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.100000e-02 VOLUME (M3)
          0.442000e-03 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  30       TYPE  19
 FLOW BALANCE CONTROL

 1      INPUTS:
        FLOW            10 - MASS FLOW RATE DETERMINED IN THIS
SUPERBLOCK
        PRESSURE        38 - PRESSURE WHERE FLOW ENTERS THIS SUPERBLOCK
        FLOW             1 - MASS FLOW RATE DETERMINED IN ANOTHER
SUPERBLOCK

 2      OUTPUTS:
        PRESSURE         7 - PRESSURE WHERE FLOW EXITS THIS SUPERBLOCK

 3      PARAMETERS:
          0.100000     INITIAL FLOW RESISTANCE PARAM. AT TIME = 0
[1000/(KG M)
 ----------------------------------------------------
```

```
UNIT  31      TYPE  76
DISCRETE-TIME HEURISTIC FAN CONTROLLER

1       INPUTS:
         CONTROL            2 - CONTROLLED VARIABLE
         CONTROL            5 - SET POINT FOR CONTROLLED VARIABLE
         CONTROL            4 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2       OUTPUTS:
         CONTROL            4 - OUTPUT CONTROL SIGNAL

3       PARAMETERS:
           0.450000e-03 INCREMENT RATE (SEC-1)
           0.450000e-03 DECREMENT RATE (SEC-1)
           0.100000e-01 CONTROLLER TIME CONSTANT (SEC)
            30.0000     SAMPLE TIME (SEC)
           0.667000     LOW LIMIT (DIMENSIONLESS)
            1.00000     HIGH LIMIT (DIMENSIONLESS)
           0.800000     INITIAL OUTPUT (DIMENSIONLESS)
           0.800000e-02 DEADBAND (DIMENSIONLESS)
----------------------------------------------------

UNIT  32      TYPE  70
FAN OR PUMP SPEED CONTROL

1       INPUTS:
         CONTROL            6 - CONTROL VARIABLE FROM CONTROLLER

2       OUTPUTS:
         RVPS               2 - FAN OR PUMP SPEED

3       PARAMETERS:
            15.0000     FAN OR PUMP SPEED WHEN CONTROL VARIABLE = 1 [RPS]
----------------------------------------------------

UNIT  33      TYPE  67
MODIFIED RATE LIMIT ACTUATOR

1       INPUTS:
         CONTROL            4 - CONTROL SIGNAL INPUT TO ACTUATOR

2       OUTPUTS:
         CONTROL            6 - ACTUATOR POSITION
         CONTROL            0 - VALVE/DAMPER POSITION
         CONTROL            0 - NUMBER OF STOP/STARTS/REVERSALS

3       PARAMETERS:
            300.000     TRAVEL TIME (LIMIT TO LIMIT) (s)
           0.390000e-02 MINIMUM CHANGE IN DEMANDED POSITION (-)
                 0.     HYSTERESIS (-)
                 0.     CRANK TRAVEL ANGLE (0 FOR LINEAR) (DEGREES)
----------------------------------------------------

UNIT  34      TYPE  15
ROOM MODEL

1       INPUTS:
```

```
        FLOW                5 - MASS FLOW RATE OF VENTILATION AIR
        TEMPERATURE        17 - VENTILATION AIR INLET TEMPERATURE
        TEMPERATURE        26 - TEMP., FULLY MIXED PART OF AIR MASS [SAME
AS OUT(1)]
        TEMPERATURE        27 - WALL MASS TEMPERATURE (SAME AS SECOND
OUTPUT)
        TEMPERATURE        28 - INTERIOR MASS TEMPERATURE (SAME AS THIRD
OUTPUT)
        TEMPERATURE        24 - SPATIAL AVG. TEMP., PISTON FLOW AIR MASS
(OUTPUT 4)
        POWER               2 - CONDUCTION HEAT FLOW INTO WALL MASS
        POWER               3 - HEAT FLOW DUE TO INTERNAL GAINS

 2     OUTPUTS:
        TEMPERATURE        26 - TEMPERATURE OF FULLY MIXED PORTION OF AIR
MASS
        TEMPERATURE        27 - WALL MASS TEMPERATURE
        TEMPERATURE        28 - INTERIOR MASS TEMPERATURE
        TEMPERATURE        24 - SPATIAL AVG. TEMP., PISTON FLOW PORTION OF
AIR MASS
        TEMPERATURE        29 - AVERAGE ROOM AIR TEMPERATURE
        TEMPERATURE        30 - EXHAUST AIR TEMPERATURE

 3     PARAMETERS:
           48.5000      VOLUME OF ROOM AIR MASS (M3)
           2000.00      THERMAL CAPACITANCE OF WALLS (KJ/C)
           4240.00      THERMAL CAPACITANCE OF INTERIOR MASS (KJ/C)
           1.00000      HEAT TRANSFER COEFF. TIMES AREA FOR WALL MASS
(KW/C)
          0.500000      HEAT TRANSFER COEFF. TIMES AREA FOR INTERIOR MASS
(KW/C
          0.300000      FRACTION OF AIR MASS WHICH IS FULLY MIXED
 ----------------------------------------------------
 Initial Variable Values:

 PRESSURE         1 ->   0.423000     (kPa)
 PRESSURE         2 ->  -0.576000e-01 (kPa)
 PRESSURE         3 ->  -0.577000e-01 (kPa)
 PRESSURE         4 ->  -0.692000e-01 (kPa)
 PRESSURE         5 ->   0.194000     (kPa)
 PRESSURE         6 ->         0.     (kPa)
 PRESSURE         7 ->   0.423000     (kPa)
 PRESSURE         8 ->  -0.576000e-01 (kPa)
 PRESSURE         9 ->   0.925000e-01 (kPa)
 PRESSURE        10 ->  -0.576000e-01 (kPa)
 PRESSURE        11 ->   0.316000e-01 (kPa)
 PRESSURE        12 ->   0.366000     (kPa)
 PRESSURE        13 ->   0.365000     (kPa)
 PRESSURE        14 ->  -0.254000     (kPa)
 PRESSURE        15 ->   0.310000e-01 (kPa)
 PRESSURE        16 ->         0.     (kPa)
 PRESSURE        17 ->   0.246000e-01 (kPa)
 PRESSURE        18 ->   0.591000e-02 (kPa)
 PRESSURE        19 ->         0.     (kPa)
 PRESSURE        20 ->         0.     (kPa)
 PRESSURE        21 ->         0.     (kPa)
 PRESSURE        22 ->   0.312000e-01 (kPa)
 PRESSURE        23 ->         0.     (kPa)
```

PRESSURE	24	->	0.	(kPa)
PRESSURE	25	->	0.	(kPa)
PRESSURE	26	->	0.	(kPa)
PRESSURE	27	->	0.	(kPa)
PRESSURE	28	->	0.	(kPa)
PRESSURE	29	->	0.	(kPa)
PRESSURE	30	->	0.	(kPa)
PRESSURE	31	->	0.365000	(kPa)
PRESSURE	32	->	0.348000	(kPa)
PRESSURE	33	->	0.282000	(kPa)
PRESSURE	34	->	0.	(kPa)
PRESSURE	35	->	0.242000	(kPa)
PRESSURE	36	->	0.427000	(kPa)
PRESSURE	37	->	0.	(kPa)
PRESSURE	38	->	0.	(kPa)
FLOW	1	->	0.427000	(kg/s)
FLOW	2	->	1.19000	(kg/s)
FLOW	3	->	0.759000	(kg/s)
FLOW	4	->	0.868000	(kg/s)
FLOW	5	->	0.283000	(kg/s)
FLOW	6	->	0.585000	(kg/s)
FLOW	7	->	0.318000	(kg/s)
FLOW	8	->	24.0000	(kg/s)
FLOW	9	->	1.59000	(kg/s)
FLOW	10	->	0.287000	(kg/s)
FLOW	11	->	1.30000	(kg/s)
FLOW	12	->	22.4000	(kg/s)
FLOW	13	->	1.30000	(kg/s)
FLOW	14	->	22.4000	(kg/s)
TEMPERATURE	1	->	14.4000	(C)
TEMPERATURE	2	->	0.	(C)
TEMPERATURE	3	->	17.2000	(C)
TEMPERATURE	4	->	17.2000	(C)
TEMPERATURE	5	->	18.1000	(C)
TEMPERATURE	6	->	19.0000	(C)
TEMPERATURE	7	->	13.3000	(C)
TEMPERATURE	8	->	18.8000	(C)
TEMPERATURE	9	->	18.1000	(C)
TEMPERATURE	10	->	0.	(C)
TEMPERATURE	11	->	0.	(C)
TEMPERATURE	12	->	10.2000	(C)
TEMPERATURE	13	->	0.	(C)
TEMPERATURE	14	->	10.0000	(C)
TEMPERATURE	15	->	0.	(C)
TEMPERATURE	16	->	0.	(C)
TEMPERATURE	17	->	18.0000	(C)
TEMPERATURE	18	->	0.	(C)
TEMPERATURE	19	->	0.	(C)
TEMPERATURE	20	->	0.	(C)
TEMPERATURE	21	->	0.	(C)
TEMPERATURE	22	->	0.	(C)
TEMPERATURE	23	->	0.	(C)
TEMPERATURE	24	->	21.6000	(C)
TEMPERATURE	25	->	21.1000	(C)
TEMPERATURE	26	->	20.5000	(C)
TEMPERATURE	27	->	23.1000	(C)
TEMPERATURE	28	->	20.1000	(C)
TEMPERATURE	29	->	21.3000	(C)

```
TEMPERATURE    30 ->      22.0000     (C)
TEMPERATURE    31 ->           0.     (C)
TEMPERATURE    32 ->      10.2000     (C)
TEMPERATURE    33 ->           0.     (C)
TEMPERATURE    34 ->           0.     (C)
TEMPERATURE    35 ->           0.     (C)
TEMPERATURE    36 ->      10.2000     (C)
TEMPERATURE    37 ->      22.5000     (C)
TEMPERATURE    38 ->      10.5000     (C)
CONTROL         1 ->     0.155000     (-)
CONTROL         2 ->     0.274000     (-)
CONTROL         3 ->     0.140000     (-)
CONTROL         4 ->     0.739000     (-)
CONTROL         5 ->     0.258000     (-)
CONTROL         6 ->     0.739000     (-)
CONTROL         7 ->           0.     (-)
CONTROL         8 ->     0.438000     (-)
CONTROL         9 ->     0.411000     (-)
RVPS            1 ->      14.8000     (rev/s)
RVPS            2 ->      11.1000     (rev/s)
POWER           1 ->      1.30000     (kW)
POWER           2 ->      2.26000     (kW)
POWER           3 ->           0.     (kW)
POWER           4 ->      17.7000     (kW)
---------------------------------------------------
Simulation Error Tolerances:

1       RTOLX=    0.100000e-04      ATOLX=    0.100000e-05
        XTOL=     0.200000e-04      TTIME=    0.100000

SUPERBLOCK 1
2       FREEZE OPTION 2       SCAN OPTION 1

SUPERBLOCK 2
3       FREEZE OPTION 2       SCAN OPTION 1

SUPERBLOCK 3
4       FREEZE OPTION 2       SCAN OPTION 1
---------------------------------------------------
The following are Boundary Variables in the simulation:

POWER            2
POWER            3
---------------------------------------------------
The following are the reported variables:


SUPERBLOCK 1      REPORTING INTERVAL    60.0000
PRESSURE         7
PRESSURE        14
PRESSURE        36
FLOW             1
FLOW             2
FLOW             3
FLOW             5
FLOW             8
TEMPERATURE      6
TEMPERATURE      7
```

TEMPERATURE	9
TEMPERATURE	24
TEMPERATURE	38
CONTROL	1
CONTROL	2
CONTROL	3
CONTROL	4
CONTROL	5
CONTROL	6
RVPS	2
POWER	2
POWER	3

SUPERBLOCK 2 REPORTING INTERVAL 20000.0

SUPERBLOCK 3 REPORTING INTERVAL 20000.0

Listing C.9. Simulation Configuration—Modified PI Fan Controller

```
HVACGEN - Simulation GENeration Program
Version 1.9B     (02-17-1987)

Fan speed minimization using modified PI controller - s1209d

SUPERBLOCK 1
     BLOCK 1
          UNIT  1      TYPE  5 - DAMPER OR VALVE
          UNIT  2      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
          UNIT  3      TYPE 72 - FLOW SENSOR
          UNIT  4      TYPE 67 - MODIFIED RATE LIMIT ACTUATOR
          UNIT  5      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT  6      TYPE  1 - FAN OR PUMP
          UNIT  7      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
          UNIT  8      TYPE  4 - FLOW MERGE
          UNIT  9      TYPE 74 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
          UNIT 10      TYPE 74 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
          UNIT 11      TYPE  7 - TEMPERATURE SENSOR
          UNIT 12      TYPE  7 - TEMPERATURE SENSOR
          UNIT 13      TYPE 28 - CONSTANT FLOW RESISTANCE
          UNIT 14      TYPE  2 - CONDUIT (DUCT OR PIPE)
     BLOCK 2
          UNIT 15      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT 16      TYPE  6 - FLOW SPLIT
          UNIT 17      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT 18      TYPE 28 - CONSTANT FLOW RESISTANCE
          UNIT 19      TYPE  6 - FLOW SPLIT
          UNIT 20      TYPE 28 - CONSTANT FLOW RESISTANCE

SUPERBLOCK 2
     BLOCK 3
          UNIT 21      TYPE  1 - FAN OR PUMP
          UNIT 22      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT 23      TYPE  6 - FLOW SPLIT
          UNIT 24      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT 25      TYPE  6 - FLOW SPLIT
          UNIT 26      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT 27      TYPE 19 - FLOW BALANCE CONTROL
          UNIT 28      TYPE 19 - FLOW BALANCE CONTROL
          UNIT 29      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
          UNIT 30      TYPE 19 - FLOW BALANCE CONTROL
          UNIT 31      TYPE 75 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
AND CO
          UNIT 32      TYPE 70 - FAN OR PUMP SPEED CONTROL
          UNIT 33      TYPE 67 - MODIFIED RATE LIMIT ACTUATOR

SUPERBLOCK 3
     BLOCK 4
          UNIT 34      TYPE 15 - ROOM MODEL
----------------------------------------------------

UNIT   1      TYPE   5
DAMPER OR VALVE

1     INPUTS:
```

```
        FLOW              1 - FLUID MASS FLOW RATE
        PRESSURE          2 - OUTLET PRESSURE
        CONTROL           1 - CONTROL: RELATIVE POSITION OF DAMPER OR
VALVE ( - )

 2     OUTPUTS:
        PRESSURE          1 - INLET PRESSURE

 3     PARAMETERS:
          0.120000     FLOW RESISTANCE, DAMPER OR VALVE OPEN [1000/(KG
M)]
          0.900000e-01 LEAKAGE PARAMETER (DIMENSIONLESS)
          0.916900     CHARACTERISTIC: 0=>EXP., 1=>LIN.,
INTERMEDIATE=>INTERME
                0.     MODE: 0=>CLOSED WHEN CONTROL=0; 1=>CLOSED WHEN
CONTROL=
 ----------------------------------------------------

 UNIT   2      TYPE   3
 INLET CONDUIT (DUCT OR PIPE)

 1     INPUTS:
        PRESSURE          7 - INLET FLUID PRESSURE
        PRESSURE          1 - OUTLET FLUID PRESSURE
        TEMPERATURE       7 - INLET FLUID TEMPERATURE
        TEMPERATURE       6 - AMBIENT AIR TEMPERATURE
        TEMPERATURE       1 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2     OUTPUTS:
        TEMPERATURE       1 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        FLOW              1 - FLUID MASS FLOW RATE

 3     PARAMETERS:
          0.390000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.410000e-03 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.484000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.185000e-02 VOLUME (M3)
          0.100000e-03 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT   3      TYPE  72
 FLOW SENSOR

 1     INPUTS:
        FLOW              1 - INPUT MASS FLOW RATE
        CONTROL           2 - FLOW SENSOR OUTPUT

 2     OUTPUTS:
        CONTROL           2 - FLOW SENSOR OUTPUT

 3     PARAMETERS:
          0.100000     SENSOR TIME CONSTANT (SEC)
                0.     FLOW OFFSET (kg/s)
```

```
           1.56000     FLOW GAIN (kg/s)
 ----------------------------------------------------

 UNIT   4       TYPE  67
 MODIFIED RATE LIMIT ACTUATOR

 1      INPUTS:
         CONTROL            3 - CONTROL SIGNAL INPUT TO ACTUATOR

 2      OUTPUTS:
         CONTROL            0 - ACTUATOR POSITION
         CONTROL            1 - VALVE/DAMPER POSITION
         CONTROL            0 - NUMBER OF STOP/STARTS/REVERSALS

 3      PARAMETERS:
           180.000     TRAVEL TIME (LIMIT TO LIMIT) (s)
          0.391000e-02 MINIMUM CHANGE IN DEMANDED POSITION (-)
                0.     HYSTERESIS (-)
           90.0000     CRANK TRAVEL ANGLE (0 FOR LINEAR) (DEGREES)
 ----------------------------------------------------

 UNIT   5       TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW               2 - FLUID MASS FLOW RATE
         PRESSURE           4 - OUTLET PRESSURE
         TEMPERATURE        3 - FLUID INLET TEMPERATURE
         TEMPERATURE        6 - AMBIENT TEMPERATURE
         TEMPERATURE        4 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE        4 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE           3 - INLET PRESSURE

 3      PARAMETERS:
          0.520000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.460000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           18.4000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.400000     VOLUME (M3)
          0.816000e-02 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT   6       TYPE   1
 FAN OR PUMP

 1      INPUTS:
         FLOW               2 - MASS FLOW RATE OF FLUID
         PRESSURE           5 - OUTLET PRESSURE
         RVPS               1 - FAN OR PUMP ROTATIONAL SPEED
         TEMPERATURE        4 - INLET FLUID TEMPERATURE

 2      OUTPUTS:
```

```
           PRESSURE          4 - INLET PRESSURE
           TEMPERATURE       5 - OUTLET FLUID TEMPERATURE
           POWER             1 - POWER CONSUMPTION

  3       PARAMETERS:
              99.1000      1ST PRESSURE COEFFICIENT
             -45.5000      2ND PRESSURE COEFFICIENT
              8.43000      3RD PRESSURE COEFFICIENT
            -0.522000      4TH PRESSURE COEFFICIENT
                   0.      5TH PRESSURE COEFFICIENT
             0.200000      1ST EFFICIENCY COEFFICIENT
                   0.      2ND EFFICIENCY COEFFICIENT
                   0.      3RD EFFICIENCY COEFFICIENT
                   0.      4TH EFFICIENCY COEFFICIENT
                   0.      5TH EFFICIENCY COEFFICIENT
             0.241000      DIAMETER (M)
              1.00000      MODE: AIR=1, WATER=2
 -----------------------------------------------------

 UNIT    7       TYPE    3
 INLET CONDUIT (DUCT OR PIPE)

  1       INPUTS:
           PRESSURE          6 - INLET FLUID PRESSURE
           PRESSURE          8 - OUTLET FLUID PRESSURE
           TEMPERATURE       6 - INLET FLUID TEMPERATURE
           TEMPERATURE       6 - AMBIENT AIR TEMPERATURE
           TEMPERATURE       8 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

  2       OUTPUTS:
           TEMPERATURE       8 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
           FLOW              3 - FLUID MASS FLOW RATE

  3       PARAMETERS:
                   0.      INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                   0.      OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                   0.      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
                   0.      VOLUME (M3)
             0.100000      FLOW RESISTANCE [1000/(KG M)]
                   0.      HEIGHT OF OUTLET ABOVE INLET (M)
              1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 -----------------------------------------------------

 UNIT    8       TYPE    4
 FLOW MERGE

  1       INPUTS:
           FLOW              3 - INLET MASS FLOW RATE 1
           FLOW              1 - INLET MASS FLOW RATE 2
           PRESSURE          3 - OUTLET PRESSURE
           TEMPERATURE       8 - INLET TEMPERATURE 1
           TEMPERATURE       1 - INLET TEMPERATURE 2

  2       OUTPUTS:
           FLOW              2 - OUTLET MASS FLOW RATE
```

```
       PRESSURE          8 - INLET PRESSURE 1
       PRESSURE         10 - INLET PRESSURE 2
       TEMPERATURE       3 - OUTLET TEMPERATURE

3      PARAMETERS:
          0.120000e-03 FLOW RESISTANCE [1000/(KG M)]
----------------------------------------------------

UNIT   9      TYPE  74
DISCRETE-TIME PI CONTROLLER W/DEADBAND

1      INPUTS:
        CONTROL           2 - CONTROLLED VARIABLE
        CONTROL           5 - SET POINT FOR CONTROLLED VARIABLE
        CONTROL           3 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2      OUTPUTS:
        CONTROL           3 - OUTPUT CONTROL SIGNAL

3      PARAMETERS:
          0.300000     PROPORTIONAL GAIN (DIMENSIONLESS)
          0.920000e-02 INTEGRAL GAIN (SEC-1)
          0.100000     CONTROLLER TIME CONSTANT (SEC)
           10.0000     SAMPLE TIME (SEC)
                0.     LOW LIMIT (DIMENSIONLESS)
           1.00000     HIGH LIMIT (DIMENSIONLESS)
          0.500000e-01 INITIAL OUTPUT (DIMENSIONLESS)
          0.391000e-02 DEADBAND (DIMENSIONLESS)
----------------------------------------------------

UNIT  10      TYPE  74
DISCRETE-TIME PI CONTROLLER W/DEADBAND

1      INPUTS:
        CONTROL           8 - CONTROLLED VARIABLE
        CONTROL           9 - SET POINT FOR CONTROLLED VARIABLE
        CONTROL           5 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2      OUTPUTS:
        CONTROL           5 - OUTPUT CONTROL SIGNAL

3      PARAMETERS:
          -11.3000     PROPORTIONAL GAIN (DIMENSIONLESS)
                0.     INTEGRAL GAIN (SEC-1)
          0.100000     CONTROLLER TIME CONSTANT (SEC)
           10.0000     SAMPLE TIME (SEC)
          0.120000     LOW LIMIT (DIMENSIONLESS)
          0.833000     HIGH LIMIT (DIMENSIONLESS)
                0.     INITIAL OUTPUT (DIMENSIONLESS)
          0.391000e-02 DEADBAND (DIMENSIONLESS)
----------------------------------------------------

UNIT  11      TYPE   7
TEMPERATURE SENSOR

1      INPUTS:
        TEMPERATURE      24 - INPUT TEMPERATURE
        CONTROL           8 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)
```

```
2      OUTPUTS:
        CONTROL            8 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

3      PARAMETERS:
          0.100000     SENSOR TIME CONSTANT (SEC)
           13.3300     TEMPERATURE OFFSET (C)
           18.8900     TEMPERATURE GAIN (C)
----------------------------------------------------

UNIT  12      TYPE   7
TEMPERATURE SENSOR

1      INPUTS:
        TEMPERATURE       25 - INPUT TEMPERATURE
        CONTROL            9 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

2      OUTPUTS:
        CONTROL            9 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

3      PARAMETERS:
          0.100000     SENSOR TIME CONSTANT (SEC)
           13.3300     TEMPERATURE OFFSET (C)
           18.8900     TEMPERATURE GAIN (C)
----------------------------------------------------

UNIT  13      TYPE  28
CONSTANT FLOW RESISTANCE

1      INPUTS:
        FLOW               1 - FLUID MASS FLOW RATE
        PRESSURE          10 - OUTLET PRESSURE

2      OUTPUTS:
        PRESSURE           2 - INLET PRESSURE

3      PARAMETERS:
          0.100000e-03 FLOW RESISTANCE  [1000/(KG M)]
----------------------------------------------------

UNIT  14      TYPE   2
CONDUIT (DUCT OR PIPE)

1      INPUTS:
        FLOW               2 - FLUID MASS FLOW RATE
        PRESSURE           9 - OUTLET PRESSURE
        TEMPERATURE        5 - FLUID INLET TEMPERATURE
        TEMPERATURE        6 - AMBIENT TEMPERATURE
        TEMPERATURE        9 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

2      OUTPUTS:
        TEMPERATURE        9 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        PRESSURE           5 - INLET PRESSURE

3      PARAMETERS:
          0.310000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
```

```
          0.280000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           11.1000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.240000     VOLUME (M3)
          0.720000e-01 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  15      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW               2 - FLUID MASS FLOW RATE
         PRESSURE          11 - OUTLET PRESSURE
         TEMPERATURE        9 - FLUID INLET TEMPERATURE
         TEMPERATURE        6 - AMBIENT TEMPERATURE
         TEMPERATURE       11 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE       11 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE           9 - INLET PRESSURE

 3      PARAMETERS:
          0.120000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.402000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           42.5300     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.489000     VOLUME (M3)
          0.433000e-01 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  16      TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW               2 - INLET MASS FLOW RATE
         PRESSURE          15 - OUTLET PRESSURE 1
         PRESSURE          22 - OUTLET PRESSURE 2

 2      OUTPUTS:
         FLOW               4 - OUTLET MASS FLOW RATE 1
         FLOW               7 - OUTLET MASS FLOW RATE 2
         PRESSURE          11 - INLET PRESSURE

 3      PARAMETERS:
          0.605000e-03 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  17      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW               4 - FLUID MASS FLOW RATE
```

```
        PRESSURE          17 - OUTLET PRESSURE
        TEMPERATURE       11 - FLUID INLET TEMPERATURE
        TEMPERATURE        6 - AMBIENT TEMPERATURE
        TEMPERATURE       17 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2     OUTPUTS:
        TEMPERATURE       17 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        PRESSURE          15 - INLET PRESSURE

 3     PARAMETERS:
          0.170000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.180000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           9.93000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.250000     VOLUME (M3)
          0.840000e-02 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  18      TYPE  28
 CONSTANT FLOW RESISTANCE

 1     INPUTS:
        FLOW               7 - FLUID MASS FLOW RATE
        PRESSURE          23 - OUTLET PRESSURE

 2     OUTPUTS:
        PRESSURE          22 - INLET PRESSURE

 3     PARAMETERS:
          0.308000     FLOW RESISTANCE  [1000/(KG M)]
 ----------------------------------------------------

 UNIT  19      TYPE   6
 FLOW SPLIT

 1     INPUTS:
        FLOW               4 - INLET MASS FLOW RATE
        PRESSURE          18 - OUTLET PRESSURE 1
        PRESSURE          20 - OUTLET PRESSURE 2

 2     OUTPUTS:
        FLOW               5 - OUTLET MASS FLOW RATE 1
        FLOW               6 - OUTLET MASS FLOW RATE 2
        PRESSURE          17 - INLET PRESSURE

 3     PARAMETERS:
          0.450000e-01 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  20      TYPE  28
 CONSTANT FLOW RESISTANCE

 1     INPUTS:
        FLOW               5 - FLUID MASS FLOW RATE
```

```
       PRESSURE          19 - OUTLET PRESSURE

 2     OUTPUTS:
       PRESSURE          18 - INLET PRESSURE

 3     PARAMETERS:
          0.739000e-01 FLOW RESISTANCE  [1000/(KG M)]
----------------------------------------------------

UNIT  21      TYPE   1
FAN OR PUMP

 1     INPUTS:
       FLOW               8 - MASS FLOW RATE OF FLUID
       PRESSURE          36 - OUTLET PRESSURE
       RVPS               2 - FAN OR PUMP ROTATIONAL SPEED
       TEMPERATURE       14 - INLET FLUID TEMPERATURE

 2     OUTPUTS:
       PRESSURE          14 - INLET PRESSURE
       TEMPERATURE       36 - OUTLET FLUID TEMPERATURE
       POWER              4 - POWER CONSUMPTION

 3     PARAMETERS:
           1.35590      1ST PRESSURE COEFFICIENT
          0.254500      2ND PRESSURE COEFFICIENT
          -2.27000      3RD PRESSURE COEFFICIENT
          -40.0060      4TH PRESSURE COEFFICIENT
                0.      5TH PRESSURE COEFFICIENT
                0.      1ST EFFICIENCY COEFFICIENT
           10.1500      2ND EFFICIENCY COEFFICIENT
          -31.9800      3RD EFFICIENCY COEFFICIENT
                0.      4TH EFFICIENCY COEFFICIENT
                0.      5TH EFFICIENCY COEFFICIENT
           2.11000      DIAMETER (M)
           1.00000      MODE: AIR=1, WATER=2
----------------------------------------------------

UNIT  22      TYPE   2
CONDUIT (DUCT OR PIPE)

 1     INPUTS:
       FLOW               8 - FLUID MASS FLOW RATE
       PRESSURE          12 - OUTLET PRESSURE
       TEMPERATURE       36 - FLUID INLET TEMPERATURE
       TEMPERATURE       37 - AMBIENT TEMPERATURE
       TEMPERATURE       12 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2     OUTPUTS:
       TEMPERATURE       12 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
       PRESSURE          36 - INLET PRESSURE

 3     PARAMETERS:
          0.500000      INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.218000      OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           1402.00      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
```

```
          49.6000      VOLUME (M3)
         0.100000e-05 FLOW RESISTANCE [1000/(KG M)]
          5.20000      HEIGHT OF OUTLET ABOVE INLET (M)
          1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  23      TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW              8 - INLET MASS FLOW RATE
         PRESSURE         31 - OUTLET PRESSURE 1
         PRESSURE         13 - OUTLET PRESSURE 2

 2      OUTPUTS:
         FLOW              9 - OUTLET MASS FLOW RATE 1
         FLOW             12 - OUTLET MASS FLOW RATE 2
         PRESSURE         12 - INLET PRESSURE

 3      PARAMETERS:
           0.100000e-05 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  24      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW              9 - FLUID MASS FLOW RATE
         PRESSURE         32 - OUTLET PRESSURE
         TEMPERATURE      12 - FLUID INLET TEMPERATURE
         TEMPERATURE      37 - AMBIENT TEMPERATURE
         TEMPERATURE      32 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE      32 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE         31 - INLET PRESSURE

 3      PARAMETERS:
           0.120000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           0.183000     OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            212.000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
            10.0000     VOLUME (M3)
           0.700000e-02 FLOW RESISTANCE [1000/(KG M)]
                 0.     HEIGHT OF OUTLET ABOVE INLET (M)
            1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  25      TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW              9 - INLET MASS FLOW RATE
         PRESSURE         33 - OUTLET PRESSURE 1
         PRESSURE         35 - OUTLET PRESSURE 2
```

```
 2      OUTPUTS:
         FLOW                10 - OUTLET MASS FLOW RATE 1
         FLOW                11 - OUTLET MASS FLOW RATE 2
         PRESSURE            32 - INLET PRESSURE

 3      PARAMETERS:
            0.500000e-01 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  26      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW                10 - FLUID MASS FLOW RATE
         PRESSURE             7 - OUTLET PRESSURE
         TEMPERATURE         32 - FLUID INLET TEMPERATURE
         TEMPERATURE          6 - AMBIENT TEMPERATURE
         TEMPERATURE          0 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE          0 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE            33 - INLET PRESSURE

 3      PARAMETERS:
             1.14000      INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            0.300000e-01 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
             50.0000      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
             1.40000      VOLUME (M3)
            0.400000e-01 FLOW RESISTANCE [1000/(KG M)]
                  0.      HEIGHT OF OUTLET ABOVE INLET (M)
             1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  27      TYPE  19
 FLOW BALANCE CONTROL

 1      INPUTS:
         FLOW                12 - MASS FLOW RATE DETERMINED IN THIS
SUPERBLOCK
         PRESSURE            38 - PRESSURE WHERE FLOW ENTERS THIS SUPERBLOCK
         FLOW                14 - MASS FLOW RATE DETERMINED IN ANOTHER
SUPERBLOCK

 2      OUTPUTS:
         PRESSURE            13 - PRESSURE WHERE FLOW EXITS THIS SUPERBLOCK

 3      PARAMETERS:
            0.130000e-02 INITIAL FLOW RESISTANCE PARAM. AT TIME = 0
[1000/(KG M)
 ----------------------------------------------------

 UNIT  28      TYPE  19
 FLOW BALANCE CONTROL
```

```
 1      INPUTS:
         FLOW               11 - MASS FLOW RATE DETERMINED IN THIS
SUPERBLOCK
         PRESSURE           38 - PRESSURE WHERE FLOW ENTERS THIS SUPERBLOCK
         FLOW               13 - MASS FLOW RATE DETERMINED IN ANOTHER
SUPERBLOCK

 2      OUTPUTS:
         PRESSURE           35 - PRESSURE WHERE FLOW EXITS THIS SUPERBLOCK

 3      PARAMETERS:
           0.240000     INITIAL FLOW RESISTANCE PARAM. AT TIME = 0
[1000/(KG M)
 -----------------------------------------------------

 UNIT  29       TYPE   3
 INLET CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         PRESSURE           38 - INLET FLUID PRESSURE
         PRESSURE           14 - OUTLET FLUID PRESSURE
         TEMPERATURE        38 - INLET FLUID TEMPERATURE
         TEMPERATURE        37 - AMBIENT AIR TEMPERATURE
         TEMPERATURE        14 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE        14 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         FLOW                8 - FLUID MASS FLOW RATE

 3      PARAMETERS:
           0.100000e-04 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                 0.     OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           0.100000e-03 THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
           0.100000e-02 VOLUME (M3)
           0.442000e-03 FLOW RESISTANCE [1000/(KG M)]
                 0.     HEIGHT OF OUTLET ABOVE INLET (M)
            1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 -----------------------------------------------------

 UNIT  30       TYPE  19
 FLOW BALANCE CONTROL

 1      INPUTS:
         FLOW               10 - MASS FLOW RATE DETERMINED IN THIS
SUPERBLOCK
         PRESSURE           38 - PRESSURE WHERE FLOW ENTERS THIS SUPERBLOCK
         FLOW                1 - MASS FLOW RATE DETERMINED IN ANOTHER
SUPERBLOCK

 2      OUTPUTS:
         PRESSURE            7 - PRESSURE WHERE FLOW EXITS THIS SUPERBLOCK

 3      PARAMETERS:
           0.100000     INITIAL FLOW RESISTANCE PARAM. AT TIME = 0
[1000/(KG M)
```

```
----------------------------------------------------

UNIT  31      TYPE  75
DISCRETE-TIME PI CONTROLLER W/DEADBAND AND CONST. DECAY

1      INPUTS:
        CONTROL           2 - CONTROLLED VARIABLE
        CONTROL           5 - SET POINT FOR CONTROLLED VARIABLE
        CONTROL           4 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2      OUTPUTS:
        CONTROL           4 - OUTPUT CONTROL SIGNAL

3      PARAMETERS:
          0.301000      PROPORTIONAL GAIN (DIMENSIONLESS)
          0.200000e-01 INTEGRAL GAIN (SEC-1)
          0.100000e-01 CONTROLLER TIME CONSTANT (SEC)
           30.0000      SAMPLE TIME (SEC)
          0.667000      LOW LIMIT (DIMENSIONLESS)
           1.00000      HIGH LIMIT (DIMENSIONLESS)
          0.800000      INITIAL OUTPUT (DIMENSIONLESS)
          0.800000e-02 DEADBAND (DIMENSIONLESS)
         -0.150000e-03 DECAY (-) OR GROWTH (+) CONSTANT (SEC-1)
----------------------------------------------------

UNIT  32      TYPE  70
FAN OR PUMP SPEED CONTROL

1      INPUTS:
        CONTROL           6 - CONTROL VARIABLE FROM CONTROLLER

2      OUTPUTS:
        RVPS              2 - FAN OR PUMP SPEED

3      PARAMETERS:
           15.0000      FAN OR PUMP SPEED WHEN CONTROL VARIABLE = 1 [RPS]
----------------------------------------------------

UNIT  33      TYPE  67
MODIFIED RATE LIMIT ACTUATOR

1      INPUTS:
        CONTROL           4 - CONTROL SIGNAL INPUT TO ACTUATOR

2      OUTPUTS:
        CONTROL           6 - ACTUATOR POSITION
        CONTROL           0 - VALVE/DAMPER POSITION
        CONTROL           0 - NUMBER OF STOP/STARTS/REVERSALS

3      PARAMETERS:
           300.000      TRAVEL TIME (LIMIT TO LIMIT) (s)
          0.390000e-02 MINIMUM CHANGE IN DEMANDED POSITION (-)
                0.      HYSTERESIS (-)
                0.      CRANK TRAVEL ANGLE (0 FOR LINEAR) (DEGREES)
----------------------------------------------------

UNIT  34      TYPE  15
ROOM MODEL
```

```
 1      INPUTS:
         FLOW                5 - MASS FLOW RATE OF VENTILATION AIR
         TEMPERATURE        17 - VENTILATION AIR INLET TEMPERATURE
         TEMPERATURE        26 - TEMP., FULLY MIXED PART OF AIR MASS [SAME
AS OUT(1)]
         TEMPERATURE        27 - WALL MASS TEMPERATURE (SAME AS SECOND
OUTPUT)
         TEMPERATURE        28 - INTERIOR MASS TEMPERATURE (SAME AS THIRD
OUTPUT)
         TEMPERATURE        24 - SPATIAL AVG. TEMP., PISTON FLOW AIR MASS
(OUTPUT 4)
         POWER               2 - CONDUCTION HEAT FLOW INTO WALL MASS
         POWER               3 - HEAT FLOW DUE TO INTERNAL GAINS

 2      OUTPUTS:
         TEMPERATURE        26 - TEMPERATURE OF FULLY MIXED PORTION OF AIR
MASS
         TEMPERATURE        27 - WALL MASS TEMPERATURE
         TEMPERATURE        28 - INTERIOR MASS TEMPERATURE
         TEMPERATURE        24 - SPATIAL AVG. TEMP., PISTON FLOW PORTION OF
AIR MASS
         TEMPERATURE        29 - AVERAGE ROOM AIR TEMPERATURE
         TEMPERATURE        30 - EXHAUST AIR TEMPERATURE

 3      PARAMETERS:
            48.5000      VOLUME OF ROOM AIR MASS (M3)
            2000.00      THERMAL CAPACITANCE OF WALLS (KJ/C)
            4240.00      THERMAL CAPACITANCE OF INTERIOR MASS (KJ/C)
            1.00000      HEAT TRANSFER COEFF. TIMES AREA FOR WALL MASS
(KW/C)
           0.500000      HEAT TRANSFER COEFF. TIMES AREA FOR INTERIOR MASS
(KW/C
           0.300000      FRACTION OF AIR MASS WHICH IS FULLY MIXED
 ----------------------------------------------------
 Initial Variable Values:

 PRESSURE         1 ->   0.423000     (kPa)
 PRESSURE         2 ->  -0.576000e-01(kPa)
 PRESSURE         3 ->  -0.577000e-01(kPa)
 PRESSURE         4 ->  -0.692000e-01(kPa)
 PRESSURE         5 ->   0.194000     (kPa)
 PRESSURE         6 ->          0.    (kPa)
 PRESSURE         7 ->   0.423000     (kPa)
 PRESSURE         8 ->  -0.576000e-01(kPa)
 PRESSURE         9 ->   0.925000e-01(kPa)
 PRESSURE        10 ->  -0.576000e-01(kPa)
 PRESSURE        11 ->   0.316000e-01(kPa)
 PRESSURE        12 ->   0.366000     (kPa)
 PRESSURE        13 ->   0.365000     (kPa)
 PRESSURE        14 ->  -0.254000     (kPa)
 PRESSURE        15 ->   0.310000e-01(kPa)
 PRESSURE        16 ->          0.    (kPa)
 PRESSURE        17 ->   0.246000e-01(kPa)
 PRESSURE        18 ->   0.591000e-02(kPa)
 PRESSURE        19 ->          0.    (kPa)
 PRESSURE        20 ->          0.    (kPa)
 PRESSURE        21 ->          0.    (kPa)
```

```
PRESSURE        22 ->   0.312000e-01 (kPa)
PRESSURE        23 ->        0.      (kPa)
PRESSURE        24 ->        0.      (kPa)
PRESSURE        25 ->        0.      (kPa)
PRESSURE        26 ->        0.      (kPa)
PRESSURE        27 ->        0.      (kPa)
PRESSURE        28 ->        0.      (kPa)
PRESSURE        29 ->        0.      (kPa)
PRESSURE        30 ->        0.      (kPa)
PRESSURE        31 ->   0.365000     (kPa)
PRESSURE        32 ->   0.348000     (kPa)
PRESSURE        33 ->   0.282000     (kPa)
PRESSURE        34 ->        0.      (kPa)
PRESSURE        35 ->   0.242000     (kPa)
PRESSURE        36 ->   0.427000     (kPa)
PRESSURE        37 ->        0.      (kPa)
PRESSURE        38 ->        0.      (kPa)
FLOW             1 ->   0.427000     (kg/s)
FLOW             2 ->    1.19000     (kg/s)
FLOW             3 ->   0.759000     (kg/s)
FLOW             4 ->   0.868000     (kg/s)
FLOW             5 ->   0.283000     (kg/s)
FLOW             6 ->   0.585000     (kg/s)
FLOW             7 ->   0.318000     (kg/s)
FLOW             8 ->    24.0000     (kg/s)
FLOW             9 ->    1.59000     (kg/s)
FLOW            10 ->   0.287000     (kg/s)
FLOW            11 ->    1.30000     (kg/s)
FLOW            12 ->    22.4000     (kg/s)
FLOW            13 ->    1.30000     (kg/s)
FLOW            14 ->    22.4000     (kg/s)
TEMPERATURE      1 ->    14.4000     (C)
TEMPERATURE      2 ->        0.      (C)
TEMPERATURE      3 ->    17.2000     (C)
TEMPERATURE      4 ->    17.2000     (C)
TEMPERATURE      5 ->    18.1000     (C)
TEMPERATURE      6 ->    19.0000     (C)
TEMPERATURE      7 ->    13.3000     (C)
TEMPERATURE      8 ->    18.8000     (C)
TEMPERATURE      9 ->    18.1000     (C)
TEMPERATURE     10 ->        0.      (C)
TEMPERATURE     11 ->        0.      (C)
TEMPERATURE     12 ->    10.2000     (C)
TEMPERATURE     13 ->        0.      (C)
TEMPERATURE     14 ->    10.0000     (C)
TEMPERATURE     15 ->        0.      (C)
TEMPERATURE     16 ->        0.      (C)
TEMPERATURE     17 ->    18.0000     (C)
TEMPERATURE     18 ->        0.      (C)
TEMPERATURE     19 ->        0.      (C)
TEMPERATURE     20 ->        0.      (C)
TEMPERATURE     21 ->        0.      (C)
TEMPERATURE     22 ->        0.      (C)
TEMPERATURE     23 ->        0.      (C)
TEMPERATURE     24 ->    21.6000     (C)
TEMPERATURE     25 ->    21.1000     (C)
TEMPERATURE     26 ->    20.5000     (C)
TEMPERATURE     27 ->    23.1000     (C)
```

```
TEMPERATURE     28 ->      20.1000     (C)
TEMPERATURE     29 ->      21.3000     (C)
TEMPERATURE     30 ->      22.0000     (C)
TEMPERATURE     31 ->           0.     (C)
TEMPERATURE     32 ->      10.2000     (C)
TEMPERATURE     33 ->           0.     (C)
TEMPERATURE     34 ->           0.     (C)
TEMPERATURE     35 ->           0.     (C)
TEMPERATURE     36 ->      10.2000     (C)
TEMPERATURE     37 ->      22.5000     (C)
TEMPERATURE     38 ->      10.5000     (C)
CONTROL          1 ->     0.155000     (-)
CONTROL          2 ->     0.274000     (-)
CONTROL          3 ->     0.140000     (-)
CONTROL          4 ->     0.739000     (-)
CONTROL          5 ->     0.258000     (-)
CONTROL          6 ->     0.739000     (-)
CONTROL          7 ->           0.     (-)
CONTROL          8 ->     0.438000     (-)
CONTROL          9 ->     0.411000     (-)
RVPS             1 ->      14.8000     (rev/s)
RVPS             2 ->      11.1000     (rev/s)
POWER            1 ->      1.30000     (kW)
POWER            2 ->      2.26000     (kW)
POWER            3 ->           0.     (kW)
POWER            4 ->      17.7000     (kW)
----------------------------------------------------
Simulation Error Tolerances:

1      RTOLX=   0.100000e-04      ATOLX=   0.100000e-05
       XTOL=    0.200000e-04      TTIME=   0.100000

SUPERBLOCK 1
2      FREEZE OPTION 2      SCAN OPTION 1

SUPERBLOCK 2
3      FREEZE OPTION 2      SCAN OPTION 1

SUPERBLOCK 3
4      FREEZE OPTION 2      SCAN OPTION 1
----------------------------------------------------
The following are Boundary Variables in the simulation:

POWER           2
POWER           3
----------------------------------------------------
The following are the reported variables:


SUPERBLOCK 1       REPORTING INTERVAL    60.0000
PRESSURE        7
PRESSURE       14
PRESSURE       36
FLOW            1
FLOW            2
FLOW            3
FLOW            5
FLOW            8
```

```
TEMPERATURE      6
TEMPERATURE      7
TEMPERATURE      9
TEMPERATURE     24
TEMPERATURE     38
CONTROL          1
CONTROL          2
CONTROL          3
CONTROL          4
CONTROL          5
CONTROL          6
RVPS             2
POWER            2
POWER            3

SUPERBLOCK 2       REPORTING INTERVAL    20000.0

SUPERBLOCK 3       REPORTING INTERVAL    20000.0
```

LISTING C.10. SIMULATION CONFIGURATION—SMALL FAN SERVING SINGLE ZONE

```
HVACGEN - Simulation GENeration Program
 Version 1.9B     (02-17-1987)


 Choose from the list below:

 CReate (SImulation,BLock,UNit)

 EDit (SImulation,UNit)

 VIew (SImulation,BLock,UNit)

 HElp

 ENd

 Selection ?


 Reading from work file....
 INITIALIZING TYPES INFORMATION...

  What part of the simulation would you like to view:

 ALl the simulation information (for documentation)

 STructure (superblock,block, and unit Information)

 VAriable initial values

 ERror tolerances, variable scan and freeze options

 BOundary variables

 REported variables

 COntinue with the previous menu


 Fan speed reset control strategy.

 SUPERBLOCK 1
      BLOCK 1
          UNIT  1     TYPE  5 - DAMPER OR VALVE
          UNIT  2     TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
          UNIT  3     TYPE 72 - FLOW SENSOR
          UNIT  4     TYPE 67 - MODIFIED RATE LIMIT ACTUATOR
          UNIT  5     TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT  6     TYPE  1 - FAN OR PUMP
          UNIT  7     TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
          UNIT  8     TYPE  4 - FLOW MERGE
          UNIT  9     TYPE 74 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
          UNIT 10     TYPE 74 - DISCRETE-TIME PI CONTROLLER W/DEADBAND
          UNIT 11     TYPE  7 - TEMPERATURE SENSOR
          UNIT 12     TYPE  7 - TEMPERATURE SENSOR
```

```
          UNIT 13      TYPE 28 - CONSTANT FLOW RESISTANCE
          UNIT 14      TYPE  2 - CONDUIT (DUCT OR PIPE)
      BLOCK 2
          UNIT 15      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT 16      TYPE  6 - FLOW SPLIT
          UNIT 17      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT 18      TYPE 28 - CONSTANT FLOW RESISTANCE
          UNIT 19      TYPE  6 - FLOW SPLIT
          UNIT 20      TYPE 28 - CONSTANT FLOW RESISTANCE

 SUPERBLOCK 2
      BLOCK 3
          UNIT 21      TYPE  1 - FAN OR PUMP
          UNIT 22      TYPE  2 - CONDUIT (DUCT OR PIPE)
          UNIT 23      TYPE  3 - INLET CONDUIT (DUCT OR PIPE)
          UNIT 24      TYPE 19 - FLOW BALANCE CONTROL
          UNIT 25      TYPE 76 - DISCRETE-TIME HEURISTIC FAN CONTROLLER
          UNIT 26      TYPE 70 - FAN OR PUMP SPEED CONTROL
          UNIT 27      TYPE 67 - MODIFIED RATE LIMIT ACTUATOR

 SUPERBLOCK 3
      BLOCK 4
          UNIT 28      TYPE 15 - ROOM MODEL
 -----------------------------------------------------

 UNIT   1      TYPE   5
 DAMPER OR VALVE

 1      INPUTS:
         FLOW                1 - FLUID MASS FLOW RATE
         PRESSURE            2 - OUTLET PRESSURE
         CONTROL             1 - CONTROL: RELATIVE POSITION OF DAMPER OR
VALVE ( - )

 2      OUTPUTS:
         PRESSURE            1 - INLET PRESSURE

 3      PARAMETERS:
            0.120000     FLOW RESISTANCE, DAMPER OR VALVE OPEN [1000/(KG
M)]
            0.900000e-01 LEAKAGE PARAMETER (DIMENSIONLESS)
            0.916900     CHARACTERISTIC: 0=>EXP., 1=>LIN.,
INTERMEDIATE=>INTERME
                  0.     MODE: 0=>CLOSED WHEN CONTROL=0; 1=>CLOSED WHEN
CONTROL=
 -----------------------------------------------------

 UNIT   2      TYPE   3
 INLET CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         PRESSURE            7 - INLET FLUID PRESSURE
         PRESSURE            1 - OUTLET FLUID PRESSURE
         TEMPERATURE         7 - INLET FLUID TEMPERATURE
         TEMPERATURE         6 - AMBIENT AIR TEMPERATURE
         TEMPERATURE         1 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)
```

```
 2      OUTPUTS:
         TEMPERATURE       1 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         FLOW              1 - FLUID MASS FLOW RATE

 3      PARAMETERS:
            0.390000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            0.410000e-03 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            0.484000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
            0.185000e-02 VOLUME (M3)
            0.100000e-03 FLOW RESISTANCE [1000/(KG M)]
                  0.     HEIGHT OF OUTLET ABOVE INLET (M)
             1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 -----------------------------------------------------

 UNIT   3      TYPE  72
 FLOW SENSOR

 1      INPUTS:
         FLOW              1 - INPUT MASS FLOW RATE
         CONTROL           2 - FLOW SENSOR OUTPUT

 2      OUTPUTS:
         CONTROL           2 - FLOW SENSOR OUTPUT

 3      PARAMETERS:
            0.100000     SENSOR TIME CONSTANT (SEC)
                  0.     FLOW OFFSET (kg/s)
             1.56000     FLOW GAIN (kg/s)
 -----------------------------------------------------

 UNIT   4      TYPE  67
 MODIFIED RATE LIMIT ACTUATOR

 1      INPUTS:
         CONTROL           3 - CONTROL SIGNAL INPUT TO ACTUATOR

 2      OUTPUTS:
         CONTROL           0 - ACTUATOR POSITION
         CONTROL           1 - VALVE/DAMPER POSITION
         CONTROL           0 - NUMBER OF STOP/STARTS/REVERSALS

 3      PARAMETERS:
             180.000     TRAVEL TIME (LIMIT TO LIMIT) (s)
            0.391000e-02 MINIMUM CHANGE IN DEMANDED POSITION (-)
                  0.     HYSTERESIS (-)
             90.0000     CRANK TRAVEL ANGLE (0 FOR LINEAR) (DEGREES)
 -----------------------------------------------------

 UNIT   5      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW              2 - FLUID MASS FLOW RATE
         PRESSURE          4 - OUTLET PRESSURE
         TEMPERATURE       3 - FLUID INLET TEMPERATURE
         TEMPERATURE       6 - AMBIENT TEMPERATURE
```

```
        TEMPERATURE       4 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
        TEMPERATURE       4 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
        PRESSURE          3 - INLET PRESSURE

 3      PARAMETERS:
           0.520000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           0.460000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            18.4000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
           0.400000     VOLUME (M3)
           0.816000e-02 FLOW RESISTANCE [1000/(KG M)]
                 0.     HEIGHT OF OUTLET ABOVE INLET (M)
            1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ---------------------------------------------------

 UNIT   6       TYPE   1
 FAN OR PUMP

 1      INPUTS:
        FLOW              2 - MASS FLOW RATE OF FLUID
        PRESSURE          5 - OUTLET PRESSURE
        RVPS              1 - FAN OR PUMP ROTATIONAL SPEED
        TEMPERATURE       4 - INLET FLUID TEMPERATURE

 2      OUTPUTS:
        PRESSURE          4 - INLET PRESSURE
        TEMPERATURE       5 - OUTLET FLUID TEMPERATURE
        POWER             1 - POWER CONSUMPTION

 3      PARAMETERS:
            99.1000     1ST PRESSURE COEFFICIENT
           -45.5000     2ND PRESSURE COEFFICIENT
            8.43000     3RD PRESSURE COEFFICIENT
          -0.522000     4TH PRESSURE COEFFICIENT
                 0.     5TH PRESSURE COEFFICIENT
           0.200000     1ST EFFICIENCY COEFFICIENT
                 0.     2ND EFFICIENCY COEFFICIENT
                 0.     3RD EFFICIENCY COEFFICIENT
                 0.     4TH EFFICIENCY COEFFICIENT
                 0.     5TH EFFICIENCY COEFFICIENT
           0.241000     DIAMETER (M)
            1.00000     MODE: AIR=1, WATER=2
 ---------------------------------------------------

 UNIT   7       TYPE   3
 INLET CONDUIT (DUCT OR PIPE)

 1      INPUTS:
        PRESSURE          6 - INLET FLUID PRESSURE
        PRESSURE          8 - OUTLET FLUID PRESSURE
        TEMPERATURE       6 - INLET FLUID TEMPERATURE
        TEMPERATURE       6 - AMBIENT AIR TEMPERATURE
        TEMPERATURE       8 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)
```

```
 2      OUTPUTS:
         TEMPERATURE        8 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         FLOW               3 - FLUID MASS FLOW RATE

 3      PARAMETERS:
                  0.      INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                  0.      OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                  0.      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
                  0.      VOLUME (M3)
            0.100000      FLOW RESISTANCE [1000/(KG M)]
                  0.      HEIGHT OF OUTLET ABOVE INLET (M)
             1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 -----------------------------------------------------

 UNIT   8      TYPE   4
 FLOW MERGE

 1      INPUTS:
         FLOW               3 - INLET MASS FLOW RATE 1
         FLOW               1 - INLET MASS FLOW RATE 2
         PRESSURE           3 - OUTLET PRESSURE
         TEMPERATURE        8 - INLET TEMPERATURE 1
         TEMPERATURE        1 - INLET TEMPERATURE 2

 2      OUTPUTS:
         FLOW               2 - OUTLET MASS FLOW RATE
         PRESSURE           8 - INLET PRESSURE 1
         PRESSURE          10 - INLET PRESSURE 2
         TEMPERATURE        3 - OUTLET TEMPERATURE

 3      PARAMETERS:
            0.120000e-03 FLOW RESISTANCE [1000/(KG M)]
 -----------------------------------------------------

 UNIT   9      TYPE  74
 DISCRETE-TIME PI CONTROLLER W/DEADBAND

 1      INPUTS:
         CONTROL            2 - CONTROLLED VARIABLE
         CONTROL            5 - SET POINT FOR CONTROLLED VARIABLE
         CONTROL            3 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

 2      OUTPUTS:
         CONTROL            3 - OUTPUT CONTROL SIGNAL

 3      PARAMETERS:
            0.300000      PROPORTIONAL GAIN (DIMENSIONLESS)
            0.920000e-02 INTEGRAL GAIN (SEC-1)
            0.100000      CONTROLLER TIME CONSTANT (SEC)
             10.0000      SAMPLE TIME (SEC)
                  0.      LOW LIMIT (DIMENSIONLESS)
             1.00000      HIGH LIMIT (DIMENSIONLESS)
            0.500000e-01 INITIAL OUTPUT (DIMENSIONLESS)
            0.391000e-02 DEADBAND (DIMENSIONLESS)
 -----------------------------------------------------
```

```
UNIT  10      TYPE  74
DISCRETE-TIME PI CONTROLLER W/DEADBAND

1      INPUTS:
        CONTROL           8 - CONTROLLED VARIABLE
        CONTROL           9 - SET POINT FOR CONTROLLED VARIABLE
        CONTROL           5 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

2      OUTPUTS:
        CONTROL           5 - OUTPUT CONTROL SIGNAL

3      PARAMETERS:
          -11.3000      PROPORTIONAL GAIN (DIMENSIONLESS)
                0.      INTEGRAL GAIN (SEC-1)
          0.100000      CONTROLLER TIME CONSTANT (SEC)
           10.0000      SAMPLE TIME (SEC)
          0.120000      LOW LIMIT (DIMENSIONLESS)
          0.833000      HIGH LIMIT (DIMENSIONLESS)
                0.      INITIAL OUTPUT (DIMENSIONLESS)
          0.391000e-02 DEADBAND (DIMENSIONLESS)
-----------------------------------------------------

UNIT  11      TYPE   7
TEMPERATURE SENSOR

1      INPUTS:
        TEMPERATURE      24 - INPUT TEMPERATURE
        CONTROL           8 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

2      OUTPUTS:
        CONTROL           8 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

3      PARAMETERS:
          0.100000      SENSOR TIME CONSTANT (SEC)
           13.3300      TEMPERATURE OFFSET (C)
           18.8900      TEMPERATURE GAIN (C)
-----------------------------------------------------

UNIT  12      TYPE   7
TEMPERATURE SENSOR

1      INPUTS:
        TEMPERATURE      25 - INPUT TEMPERATURE
        CONTROL           9 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

2      OUTPUTS:
        CONTROL           9 - SENSOR OUTPUT (MODIFIED BY GAIN AND OFFSET)

3      PARAMETERS:
          0.100000      SENSOR TIME CONSTANT (SEC)
           13.3300      TEMPERATURE OFFSET (C)
           18.8900      TEMPERATURE GAIN (C)
-----------------------------------------------------

UNIT  13      TYPE  28
CONSTANT FLOW RESISTANCE
```

```
 1      INPUTS:
         FLOW                 1 - FLUID MASS FLOW RATE
         PRESSURE            10 - OUTLET PRESSURE

 2      OUTPUTS:
         PRESSURE             2 - INLET PRESSURE

 3      PARAMETERS:
            0.100000e-03 FLOW RESISTANCE  [1000/(KG M)]
 ----------------------------------------------------

 UNIT  14      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW                 2 - FLUID MASS FLOW RATE
         PRESSURE             9 - OUTLET PRESSURE
         TEMPERATURE          5 - FLUID INLET TEMPERATURE
         TEMPERATURE          6 - AMBIENT TEMPERATURE
         TEMPERATURE          9 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE          9 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE             5 - INLET PRESSURE

 3      PARAMETERS:
            0.310000e-02 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            0.280000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
             11.1000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
            0.240000     VOLUME (M3)
            0.720000e-01 FLOW RESISTANCE [1000/(KG M)]
                  0.     HEIGHT OF OUTLET ABOVE INLET (M)
             1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  15      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW                 2 - FLUID MASS FLOW RATE
         PRESSURE            11 - OUTLET PRESSURE
         TEMPERATURE          9 - FLUID INLET TEMPERATURE
         TEMPERATURE          6 - AMBIENT TEMPERATURE
         TEMPERATURE         11 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE         11 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE             9 - INLET PRESSURE

 3      PARAMETERS:
            0.120000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
            0.402000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
             42.5300     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
```

```
          0.489000     VOLUME (M3)
          0.433000e-01 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  16      TYPE   6
 FLOW SPLIT

 1      INPUTS:
         FLOW               2 - INLET MASS FLOW RATE
         PRESSURE          15 - OUTLET PRESSURE 1
         PRESSURE          22 - OUTLET PRESSURE 2

 2      OUTPUTS:
         FLOW               4 - OUTLET MASS FLOW RATE 1
         FLOW               7 - OUTLET MASS FLOW RATE 2
         PRESSURE          11 - INLET PRESSURE

 3      PARAMETERS:
          0.605000e-03 FLOW RESISTANCE [1000/(KG M)]
 ----------------------------------------------------

 UNIT  17      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW               4 - FLUID MASS FLOW RATE
         PRESSURE          17 - OUTLET PRESSURE
         TEMPERATURE       11 - FLUID INLET TEMPERATURE
         TEMPERATURE        6 - AMBIENT TEMPERATURE
         TEMPERATURE       17 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE       17 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE          15 - INLET PRESSURE

 3      PARAMETERS:
          0.170000e-01 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
          0.180000e-02 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           9.93000     THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
          0.250000     VOLUME (M3)
          0.840000e-02 FLOW RESISTANCE [1000/(KG M)]
                0.     HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000     MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  18      TYPE  28
 CONSTANT FLOW RESISTANCE

 1      INPUTS:
         FLOW               7 - FLUID MASS FLOW RATE
         PRESSURE          23 - OUTLET PRESSURE
```

```
2       OUTPUTS:
         PRESSURE          22 - INLET PRESSURE

3       PARAMETERS:
           0.308000     FLOW RESISTANCE   [1000/(KG M)]
----------------------------------------------------

UNIT  19       TYPE   6
FLOW SPLIT

1       INPUTS:
         FLOW               4 - INLET MASS FLOW RATE
         PRESSURE          18 - OUTLET PRESSURE 1
         PRESSURE          20 - OUTLET PRESSURE 2

2       OUTPUTS:
         FLOW               5 - OUTLET MASS FLOW RATE 1
         FLOW               6 - OUTLET MASS FLOW RATE 2
         PRESSURE          17 - INLET PRESSURE

3       PARAMETERS:
           0.450000e-01 FLOW RESISTANCE [1000/(KG M)]
----------------------------------------------------

UNIT  20       TYPE  28
CONSTANT FLOW RESISTANCE

1       INPUTS:
         FLOW               5 - FLUID MASS FLOW RATE
         PRESSURE          19 - OUTLET PRESSURE

2       OUTPUTS:
         PRESSURE          18 - INLET PRESSURE

3       PARAMETERS:
           0.739000e-01 FLOW RESISTANCE  [1000/(KG M)]
----------------------------------------------------

UNIT  21       TYPE   1
FAN OR PUMP

1       INPUTS:
         FLOW               8 - MASS FLOW RATE OF FLUID
         PRESSURE          36 - OUTLET PRESSURE
         RVPS               2 - FAN OR PUMP ROTATIONAL SPEED
         TEMPERATURE       14 - INLET FLUID TEMPERATURE

2       OUTPUTS:
         PRESSURE          14 - INLET PRESSURE
         TEMPERATURE       36 - OUTLET FLUID TEMPERATURE
         POWER              4 - POWER CONSUMPTION

3       PARAMETERS:
            1.35590     1ST PRESSURE COEFFICIENT
           0.254500     2ND PRESSURE COEFFICIENT
           -2.27000     3RD PRESSURE COEFFICIENT
           -40.0060     4TH PRESSURE COEFFICIENT
                 0.     5TH PRESSURE COEFFICIENT
```

```
              0.      1ST EFFICIENCY COEFFICIENT
         10.1500      2ND EFFICIENCY COEFFICIENT
        -31.9800      3RD EFFICIENCY COEFFICIENT
              0.      4TH EFFICIENCY COEFFICIENT
              0.      5TH EFFICIENCY COEFFICIENT
         1.79000      DIAMETER (M)
         1.00000      MODE: AIR=1, WATER=2
 ----------------------------------------------------

 UNIT  22      TYPE   2
 CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         FLOW                8 - FLUID MASS FLOW RATE
         PRESSURE            7 - OUTLET PRESSURE
         TEMPERATURE        36 - FLUID INLET TEMPERATURE
         TEMPERATURE         6 - AMBIENT TEMPERATURE
         TEMPERATURE         7 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE         7 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         PRESSURE           36 - INLET PRESSURE

 3      PARAMETERS:
           1.14000      INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
         0.300000e-01 OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
           50.0000      THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
           1.40000      VOLUME (M3)
         0.200000       FLOW RESISTANCE [1000/(KG M)]
                0.      HEIGHT OF OUTLET ABOVE INLET (M)
           1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  23      TYPE   3
 INLET CONDUIT (DUCT OR PIPE)

 1      INPUTS:
         PRESSURE           38 - INLET FLUID PRESSURE
         PRESSURE           14 - OUTLET FLUID PRESSURE
         TEMPERATURE        38 - INLET FLUID TEMPERATURE
         TEMPERATURE         6 - AMBIENT AIR TEMPERATURE
         TEMPERATURE        14 - OUTLET FLUID TEMPERATURE (SAME AS FIRST
OUTPUT)

 2      OUTPUTS:
         TEMPERATURE        14 - OUTLET FLUID TEMPERATURE (SAME AS FIFTH
INPUT)
         FLOW                8 - FLUID MASS FLOW RATE

 3      PARAMETERS:
         0.100000e-04 INSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
                0.    OUTSIDE HEAT TRANSFER COEFFICIENT X AREA (KW/C)
         0.100000e-03 THERMAL CAPACITANCE OF CONDUIT MATERIAL (KJ/C)
         0.100000e-02 VOLUME (M3)
         0.120000     FLOW RESISTANCE [1000/(KG M)]
```

```
              0.      HEIGHT OF OUTLET ABOVE INLET (M)
         1.00000      MODE: 2=WATER, 1=AIR, NEG.=DETAILED, POS.=SIMPLE
DYNAMI
 ----------------------------------------------------

 UNIT  24      TYPE  19
 FLOW BALANCE CONTROL

 1      INPUTS:
         FLOW               8 - MASS FLOW RATE DETERMINED IN THIS
SUPERBLOCK
         PRESSURE          38 - PRESSURE WHERE FLOW ENTERS THIS SUPERBLOCK
         FLOW               1 - MASS FLOW RATE DETERMINED IN ANOTHER
SUPERBLOCK

 2      OUTPUTS:
         PRESSURE           7 - PRESSURE WHERE FLOW EXITS THIS SUPERBLOCK

 3      PARAMETERS:
           0.100000     INITIAL FLOW RESISTANCE PARAM. AT TIME = 0
[1000/(KG M)
 ----------------------------------------------------

 UNIT  25      TYPE  76
 DISCRETE-TIME HEURISTIC FAN CONTROLLER

 1      INPUTS:
         CONTROL            2 - CONTROLLED VARIABLE
         CONTROL            5 - SET POINT FOR CONTROLLED VARIABLE
         CONTROL            4 - OUTPUT CONTROL SIGNAL (SAME AS OUTPUT)

 2      OUTPUTS:
         CONTROL            4 - OUTPUT CONTROL SIGNAL

 3      PARAMETERS:
           0.300000e-03 INCREMENT RATE (SEC-1)
           0.300000e-03 DECREMENT RATE (SEC-1)
           0.100000e-01 CONTROLLER TIME CONSTANT (SEC)
            30.0000     SAMPLE TIME (SEC)
           0.533000     LOW LIMIT (DIMENSIONLESS)
            1.00000     HIGH LIMIT (DIMENSIONLESS)
           0.800000     INITIAL OUTPUT (DIMENSIONLESS)
           0.800000e-02 DEADBAND (DIMENSIONLESS)
 ----------------------------------------------------

 UNIT  26      TYPE  70
 FAN OR PUMP SPEED CONTROL

 1      INPUTS:
         CONTROL            6 - CONTROL VARIABLE FROM CONTROLLER

 2      OUTPUTS:
         RVPS               2 - FAN OR PUMP SPEED

 3      PARAMETERS:
            15.0000     FAN OR PUMP SPEED WHEN CONTROL VARIABLE = 1 [RPS]
 ----------------------------------------------------
```

```
UNIT  27      TYPE  67
MODIFIED RATE LIMIT ACTUATOR

1      INPUTS:
        CONTROL          4 - CONTROL SIGNAL INPUT TO ACTUATOR

2      OUTPUTS:
        CONTROL          6 - ACTUATOR POSITION
        CONTROL          0 - VALVE/DAMPER POSITION
        CONTROL          0 - NUMBER OF STOP/STARTS/REVERSALS

3      PARAMETERS:
           300.000     TRAVEL TIME (LIMIT TO LIMIT) (s)
          0.390000e-02 MINIMUM CHANGE IN DEMANDED POSITION (-)
                0.     HYSTERESIS (-)
                0.     CRANK TRAVEL ANGLE (0 FOR LINEAR) (DEGREES)
----------------------------------------------------

UNIT  28      TYPE  15
ROOM MODEL

1      INPUTS:
        FLOW             5 - MASS FLOW RATE OF VENTILATION AIR
        TEMPERATURE     17 - VENTILATION AIR INLET TEMPERATURE
        TEMPERATURE     26 - TEMP., FULLY MIXED PART OF AIR MASS [SAME
AS OUT(1)]
        TEMPERATURE     27 - WALL MASS TEMPERATURE (SAME AS SECOND
OUTPUT)
        TEMPERATURE     28 - INTERIOR MASS TEMPERATURE (SAME AS THIRD
OUTPUT)
        TEMPERATURE     24 - SPATIAL AVG. TEMP., PISTON FLOW AIR MASS
(OUTPUT 4)
        POWER            2 - CONDUCTION HEAT FLOW INTO WALL MASS
        POWER            3 - HEAT FLOW DUE TO INTERNAL GAINS

2      OUTPUTS:
        TEMPERATURE     26 - TEMPERATURE OF FULLY MIXED PORTION OF AIR
MASS
        TEMPERATURE     27 - WALL MASS TEMPERATURE
        TEMPERATURE     28 - INTERIOR MASS TEMPERATURE
        TEMPERATURE     24 - SPATIAL AVG. TEMP., PISTON FLOW PORTION OF
AIR MASS
        TEMPERATURE     29 - AVERAGE ROOM AIR TEMPERATURE
        TEMPERATURE     30 - EXHAUST AIR TEMPERATURE

3      PARAMETERS:
           48.5000     VOLUME OF ROOM AIR MASS (M3)
           2000.00     THERMAL CAPACITANCE OF WALLS (KJ/C)
           4240.00     THERMAL CAPACITANCE OF INTERIOR MASS (KJ/C)
           1.00000     HEAT TRANSFER COEFF. TIMES AREA FOR WALL MASS
(KW/C)
          0.500000     HEAT TRANSFER COEFF. TIMES AREA FOR INTERIOR MASS
(KW/C
          0.300000     FRACTION OF AIR MASS WHICH IS FULLY MIXED
----------------------------------------------------
Initial Variable Values:

PRESSURE        1 ->    0.423000    (kPa)
```

```
PRESSURE          2 ->  -0.576000e-01(kPa)
PRESSURE          3 ->  -0.577000e-01(kPa)
PRESSURE          4 ->  -0.692000e-01(kPa)
PRESSURE          5 ->   0.194000     (kPa)
PRESSURE          6 ->         0.     (kPa)
PRESSURE          7 ->   0.423000     (kPa)
PRESSURE          8 ->  -0.576000e-01(kPa)
PRESSURE          9 ->   0.925000e-01(kPa)
PRESSURE         10 ->  -0.576000e-01(kPa)
PRESSURE         11 ->   0.316000e-01(kPa)
PRESSURE         12 ->   0.366000     (kPa)
PRESSURE         13 ->   0.365000     (kPa)
PRESSURE         14 ->  -0.254000     (kPa)
PRESSURE         15 ->   0.310000e-01(kPa)
PRESSURE         16 ->         0.     (kPa)
PRESSURE         17 ->   0.246000e-01(kPa)
PRESSURE         18 ->   0.591000e-02(kPa)
PRESSURE         19 ->         0.     (kPa)
PRESSURE         20 ->         0.     (kPa)
PRESSURE         21 ->         0.     (kPa)
PRESSURE         22 ->   0.312000e-01(kPa)
PRESSURE         23 ->         0.     (kPa)
PRESSURE         24 ->         0.     (kPa)
PRESSURE         25 ->         0.     (kPa)
PRESSURE         26 ->         0.     (kPa)
PRESSURE         27 ->         0.     (kPa)
PRESSURE         28 ->         0.     (kPa)
PRESSURE         29 ->         0.     (kPa)
PRESSURE         30 ->         0.     (kPa)
PRESSURE         31 ->   0.365000     (kPa)
PRESSURE         32 ->   0.348000     (kPa)
PRESSURE         33 ->   0.282000     (kPa)
PRESSURE         34 ->         0.     (kPa)
PRESSURE         35 ->   0.242000     (kPa)
PRESSURE         36 ->   0.427000     (kPa)
PRESSURE         37 ->         0.     (kPa)
PRESSURE         38 ->         0.     (kPa)
FLOW              1 ->   0.427000     (kg/s)
FLOW              2 ->    1.19000     (kg/s)
FLOW              3 ->   0.759000     (kg/s)
FLOW              4 ->   0.868000     (kg/s)
FLOW              5 ->   0.283000     (kg/s)
FLOW              6 ->   0.585000     (kg/s)
FLOW              7 ->   0.318000     (kg/s)
FLOW              8 ->    24.0000     (kg/s)
TEMPERATURE       1 ->    14.4000     (C)
TEMPERATURE       2 ->         0.     (C)
TEMPERATURE       3 ->    17.2000     (C)
TEMPERATURE       4 ->    17.2000     (C)
TEMPERATURE       5 ->    18.1000     (C)
TEMPERATURE       6 ->    19.0000     (C)
TEMPERATURE       7 ->    14.4000     (C)
TEMPERATURE       8 ->    18.8000     (C)
TEMPERATURE       9 ->    18.1000     (C)
TEMPERATURE      10 ->         0.     (C)
TEMPERATURE      11 ->         0.     (C)
TEMPERATURE      12 ->    10.2000     (C)
TEMPERATURE      13 ->         0.     (C)
```

```
TEMPERATURE     14 ->     10.0000     (C)
TEMPERATURE     15 ->          0.     (C)
TEMPERATURE     16 ->          0.     (C)
TEMPERATURE     17 ->     18.0000     (C)
TEMPERATURE     18 ->          0.     (C)
TEMPERATURE     19 ->          0.     (C)
TEMPERATURE     20 ->          0.     (C)
TEMPERATURE     21 ->          0.     (C)
TEMPERATURE     22 ->          0.     (C)
TEMPERATURE     23 ->          0.     (C)
TEMPERATURE     24 ->     21.6000     (C)
TEMPERATURE     25 ->     21.1000     (C)
TEMPERATURE     26 ->     20.5000     (C)
TEMPERATURE     27 ->     23.1000     (C)
TEMPERATURE     28 ->     20.1000     (C)
TEMPERATURE     29 ->     21.3000     (C)
TEMPERATURE     30 ->     22.0000     (C)
TEMPERATURE     31 ->          0.     (C)
TEMPERATURE     32 ->     10.2000     (C)
TEMPERATURE     33 ->          0.     (C)
TEMPERATURE     34 ->          0.     (C)
TEMPERATURE     35 ->          0.     (C)
TEMPERATURE     36 ->     10.2000     (C)
TEMPERATURE     37 ->     22.5000     (C)
TEMPERATURE     38 ->     10.5000     (C)
CONTROL          1 ->    0.155000     (-)
CONTROL          2 ->    0.274000     (-)
CONTROL          3 ->    0.140000     (-)
CONTROL          4 ->    0.739000     (-)
CONTROL          5 ->    0.258000     (-)
CONTROL          6 ->    0.739000     (-)
CONTROL          7 ->          0.     (-)
CONTROL          8 ->    0.438000     (-)
CONTROL          9 ->    0.411000     (-)
RVPS             1 ->     14.8000     (rev/s)
RVPS             2 ->     11.1000     (rev/s)
POWER            1 ->     1.30000     (kW)
POWER            2 ->     2.26000     (kW)
POWER            3 ->          0.     (kW)
POWER            4 ->     17.7000     (kW)
----------------------------------------------------
Simulation Error Tolerances:

1      RTOLX=   0.100000e-03      ATOLX=   0.100000e-04
       XTOL=    0.200000e-03      TTIME=   0.100000

SUPERBLOCK 1
2      FREEZE OPTION 2      SCAN OPTION 1

SUPERBLOCK 2
3      FREEZE OPTION 2      SCAN OPTION 1

SUPERBLOCK 3
4      FREEZE OPTION 0      SCAN OPTION 0
----------------------------------------------------
The following are Boundary Variables in the simulation:

POWER           2
```

```
POWER           3
-----------------------------------------------------
The following are the reported variables:


SUPERBLOCK 1        REPORTING INTERVAL    1.00000
PRESSURE        7
PRESSURE       14
PRESSURE       36
FLOW            1
FLOW            2
FLOW            3
FLOW            5
FLOW            8
TEMPERATURE     6
TEMPERATURE     7
TEMPERATURE     9
TEMPERATURE    24
TEMPERATURE    38
CONTROL         1
CONTROL         2
CONTROL         3
CONTROL         4
CONTROL         5
CONTROL         6
RVPS            2
POWER           2
POWER           3

SUPERBLOCK 2        REPORTING INTERVAL    20000.0

SUPERBLOCK 3        REPORTING INTERVAL    20000.0
```

This thesis was produced entirely on an Apple Macintosh, and typeset in Times Roman, Helvetica, Bookman, and Courier by the author, using a QMS PS 800-II laser printer driven by a Macintosh IIcx. Microsoft *Word* was used to integrate text with graphs produced using *Igor*, by WaveMetrics, and drawings done with *MacDraw II*, by Claris. Equations were done with *MathType*, by Design Science. *Igor* and Fox Software's *FoxBASE* were used for much of the analysis.